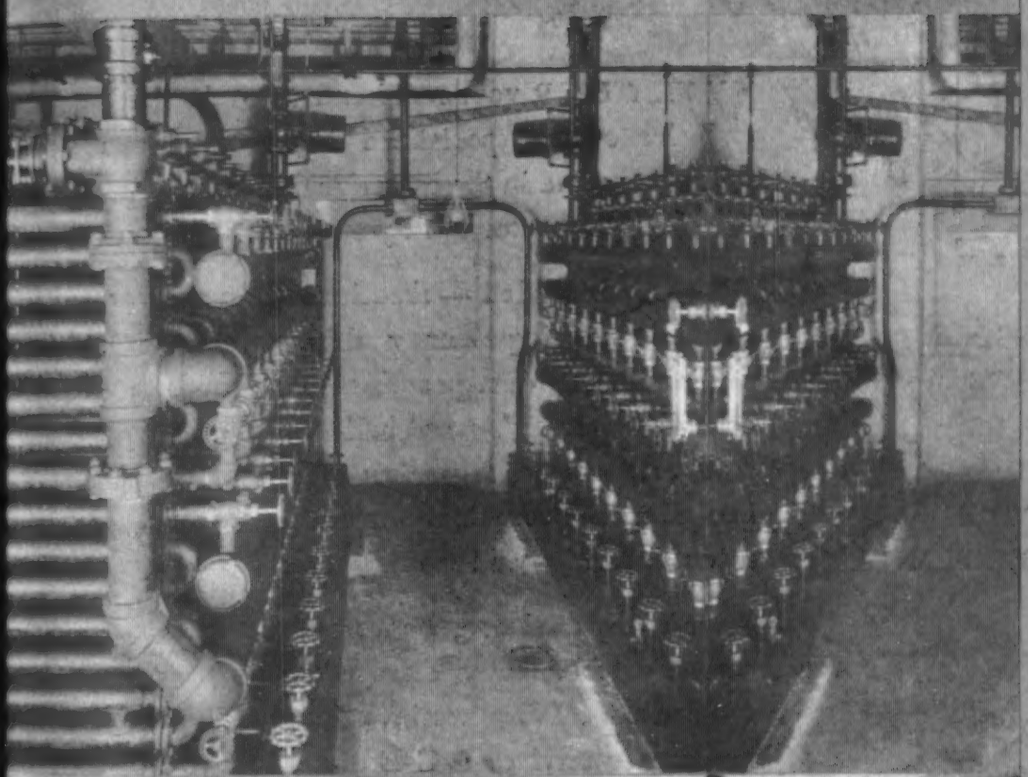
JUN 3 1931

Heating · Piping and Air Conditioning



Articles by

F. E. GIESECKE

RUSH D. TOUTON

E. C. GASTON

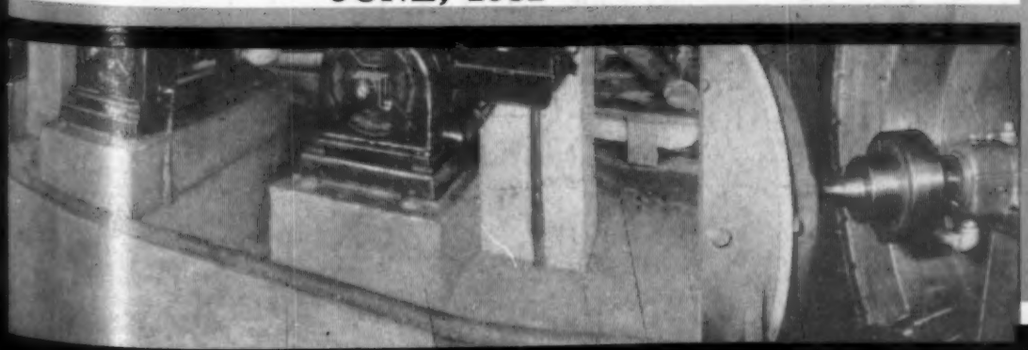
O. W. KOTHE

MALCOLM TOMLINSON

EDWARD F. ADOLPH

and others

JUNE, 1931



Journal of the
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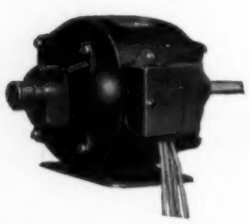

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GENERAL ELECTRIC

The EDITOR'S PAGE

Heating · Piping and Air Conditioning

FROM ARTICLES IN THIS ISSUE

"Two hundred thousand square feet of 2-inch corkboard and other unusual features in building design provide an effective barrier between the fluctuations in outside weather and the exact (air) conditions required for fine cigar making."—*Rush D. Touton.*

"It can be well stated that a steam plant consists principally of a boiler, turbo-generator, a few pumps and heaters, and PIPING—not the least of which is piping. This article will be devoted entirely to the description of the actual piping, and to the discussion of the major problems encountered in the design."—*E. C. Gaston.*

"The selection of a suitable form of drying must depend considerably on the temperature which the substance to be dried can stand without deterioration in structure, color, or appearance."—*Malcolm Tomlinson.*

"To keep the return lines and traps of a large heating system, covering an extensive territory, in proper working condition requires close inspection, frequent testing, and a generous amount of work."—*W. H. Wilson.*

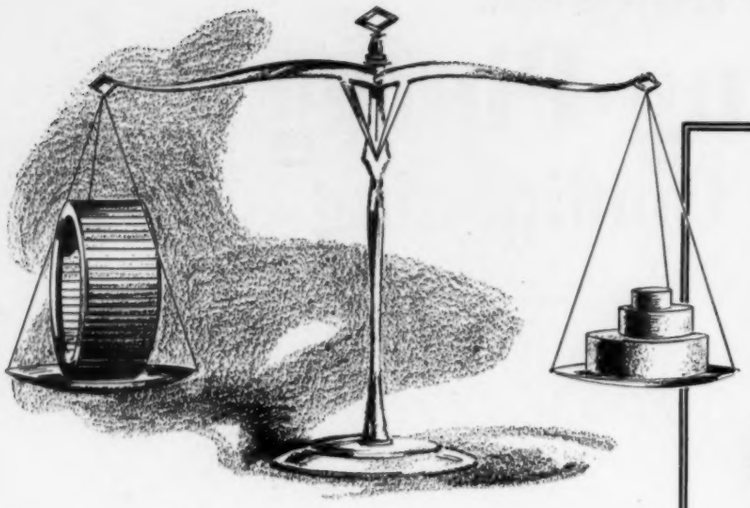
"In computing the heating surface of the pipe coil, it is generally customary to take the inner circumference, as that is the real heating surface to which heat is supplied."—*Rollo E. Gilmore.*

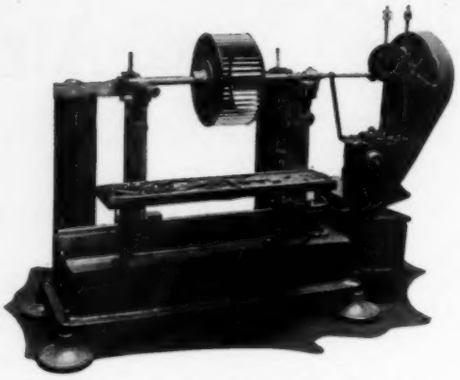
"In order to form an opinion as to the tolerance which may be permitted in a series of radiator circuits in a forced circulation system, without interfering appreciably with the satisfactory operation of the system, let us examine Fig. 6."—*F. E. Giesecke.*

"Effective temperatures are a true basis of reference for atmospheric conditions; but only insofar as they influence the human body through the temperature sense-spots of the skin." *Edward F. Adolph.*

FROM A LETTER

"Since its publication I have sent you no word of comment as to my impressions of it, and at this time I take great pleasure in doing so. In my experience with technical and other magazines I know of none that began its activities with such a pretentious program as did Heating, Piping and Air Conditioning, and what has been still better, you have more than made the objective from every point of view. The articles cover a wide range of the heating, piping and air conditioning industry; they are authoritative, based on sound theory and practice, and they are always forward looking for constructive future development."





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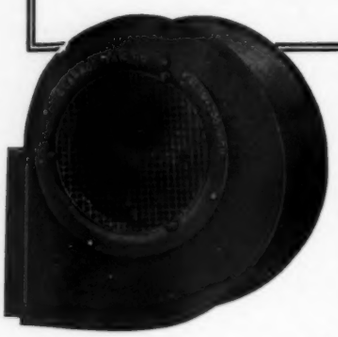

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A Clarage Fan can be furnished on any type or size of stoker installation, and it pays to insist on Clarage. Write for Illustrated Bulletin 541-A and learn all of the reasons why. **Clarage Fan Company, Kalamazoo, Michigan.**

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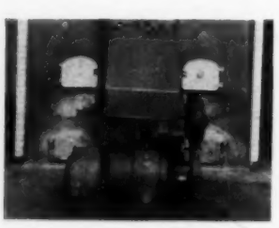
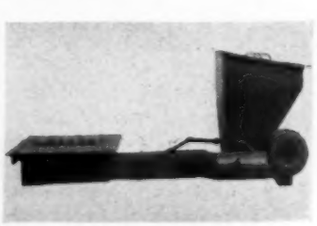


Clarage Fan and Canton Unit Stoker.

Clarage Fan and Brownell Unit Stoker.



Clarage Fan and Kol-Master Unit Stoker.

CLARAGE

UNIT STOKER FANS

BLOWERS — AIR CONDITIONING EQUIPMENT — EXHAUST FANS — ENGINES

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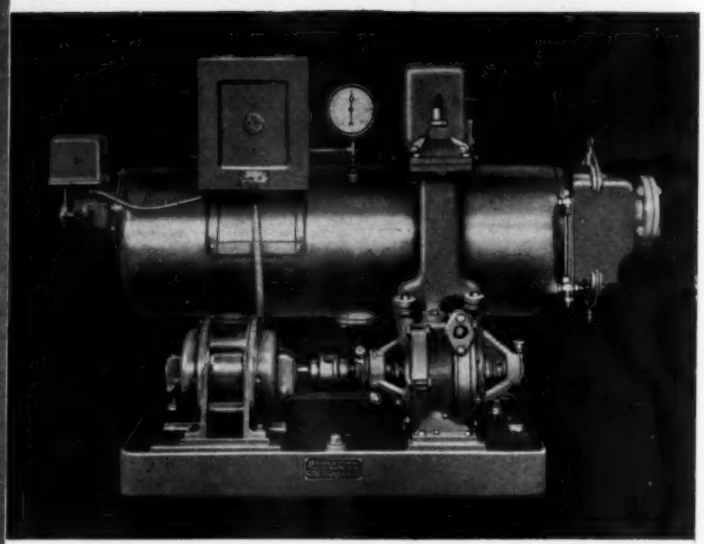

STARTING up a return line heating system in the morning throws a heavy load on the heating pump. Steam condenses rapidly in the cold radiators. Large quantities of water that have collected over night must be removed.

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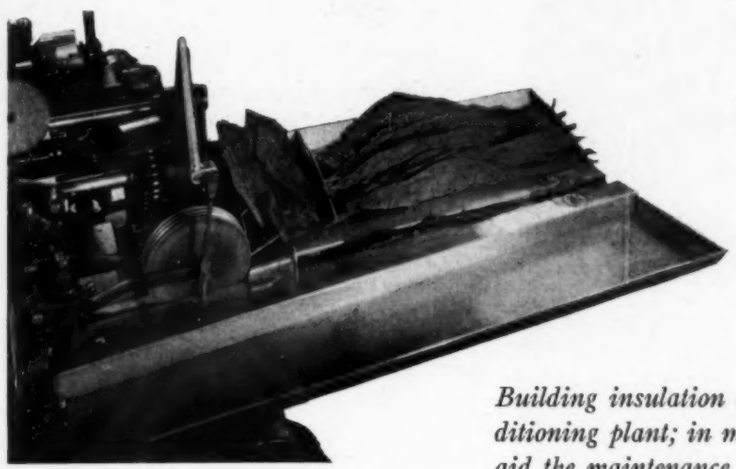
Heating · Piping and Air Conditioning

Vol. III June, 1931 No. 6



PREPARED TOBACCO ON THE FEEDING END OF THE CIGAR MACHINE. AT THE TIME OF DELIVERY TO THIS POINT IN THE PROCESS, THE TOBACCO HAS EXACTLY THE RIGHT MOISTURE CONTENT. FAULTY WALL OR WINDOW CONSTRUCTION WOULD MAKE THIS CONDITION DIFFICULT TO PRESERVE DURING THE MANUFACTURE OF THE CIGAR

By Rush D. Touton*

Building insulation can be considered a part of the air conditioning plant; in many cases, a very important part. It will aid the maintenance of proper atmospheric conditions for industrial processes, enabling close control. It will reduce the necessary investment in refrigerating equipment, and cut down the fuel and power consumption. While the needs of your plant or building may not be as exacting as those of Bayuk Cigars Inc., study of Mr. Touton's article shows the importance of careful insulation as it concerns air conditioning.

Insulation Decreases Load on Industrial Air Conditioning Plant by One-third

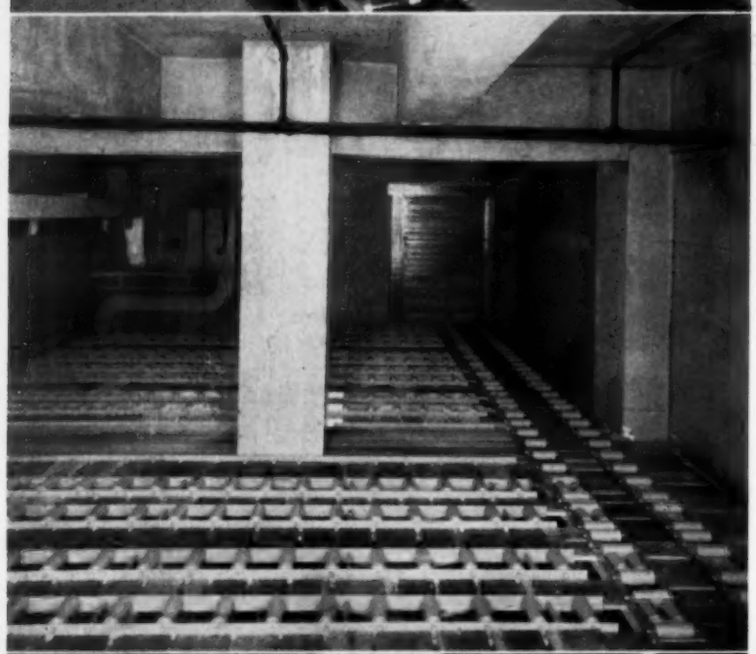
COMPLETE insulation on the walls and roof of the central plant of Bayuk Cigars Inc.[1] in Philadelphia has been largely responsible for giving it the unique name "The Giant Humidor." 200,000 sq. ft. of 2-in. corkboard and other unusual features in building design provide an effective barrier between the fluctuations in outside weather and the exact conditions required for fine cigar making.

To understand the need for this thorough thermal protection it is necessary to appreciate the hygroscopic nature of all tobaccos. The sensitivity to atmospheric environment is more pronounced in the better grades of tobacco leaf, and the moisture content of the most delicate and desirable leaves must be closely guarded. A reduction of only 2 per cent in the relative humidity will soon make the prepared leaves crisp and unworkable while an increase of the same amount will also make the stock unfit for use. This situation was taken care of in the old style cigar factories by keeping all prepared lots under moistened covers and re-covering the supply immediately after the material for one cigar was withdrawn. Such a practice could, of course, be of benefit only during the cooler months when the moisture content of the air is low.

An air conditioning system designed amply large to maintain the requirements in the ordinary type of build-

*Technical Director, Bayuk Cigars, Inc., Philadelphia, Pa.

[1] The air conditioning system in this plant was described in the July, 1930, HEATING, PIPING AND AIR CONDITIONING (Vol. II, No. 7, p. 547).



TOP—A VIEW OF THE SELECTING DEPARTMENT ON THE EIGHTH FLOOR. CIGARS ARE SUBJECTED TO A SPECIAL CONDITIONING PROCESS AND THEN ARE SELECTED. NOTE THAT DELIVERY LINES CAN BE LOCATED IMMEDIATELY ADJACENT TO EXTERIOR WALLS. MIDDLE—INTERIOR (UPPER DECK) OF FERMENTATION ROOM; NOTE INSULATED WALLS AND CEILING, AND INSULATED DOOR. BOTTOM—TWO FERMENTATION ROOMS. WALL CONSTRUCTION IS OF 6-IN. TILE AND 2-IN. CORKBOARD; BOTH SIDES FINISHED

ing would be of great size and of almost prohibitive cost.

Insulation Aids Close Control

In the case of this plant, the load on the conditioning system would have been increased by 30 per cent had the special building construction not been used. The close control now obtained, which in some processes is held to a variation of 1/10 of 1 F, would have been more difficult because of the large quantities of steam, air, etc. to be regulated. The air conditioning equipment in all departments of the insulated Bayuk plant has demonstrated both economy of operation and satisfactory performance. Two central systems furnish the 70 per cent relative humidity required for the manufacturing areas, and fifteen individual installations provide the proper climate for process work.

Advantages of the Conditioning System

Here one entire floor is devoted to the storage of over 1,000,000 pounds of leaf tobacco. This material is piled in bins, with the sides and tops exposed, for a special mellowing process, and the obviously precise conditioning required is easily maintained. The 200 fresh work cigar machines, each producing 1,000,000 cigars per year, are able to function without constant readjustment to compensate for changing moisture content of the tobacco. The sorting, banding and cellophaning processes, all exposing the materials to the air, are kept at high efficiency. All forms of leaf tobacco and the finished products are handled on an automatic conveyor system operating through the eight floors of the building. Naturally, attempting to cover materials to preserve moisture content while in transit between departments would involve considerable expense and be a certain source of trouble.

Application of the Insulation

This factory is of reinforced concrete construction with 12-in. brick walls and a 6-in. concrete roof. All smooth concrete surfaces were roughened to secure the proper bond between the corkboard and the surface. Particular care was taken to insure a tight fit

of the corkboard into all corners and against the floors and ceilings. After the corkboard had thoroughly set in place, a ¼-in. scratch coat of cement plaster was applied, followed by a second ¼-in. coat of the same material, which was then brought to a smooth trowel finish.

The coefficient of heat transmission for this type of wall is 0.097 Btu per sq. ft. per hr. per 1 F temperature difference. All figures used in this paper are based on an outside wind velocity of 15 miles per hour, and are taken from the *A.S.H.V.E. Guide*.

Two-inch corkboard was laid in hot asphalt on the roof deck with built-up roofing applied over the cork. The coefficient of transmission for this construction is 0.115 Btu per sq. ft. per hr. per 1 F difference.

Steam Pipe Prevents Condensation

All windows are double with a 6-in. air space between the sash. This feature provides a saving of 60 per cent in heat transmission through the glass since the coefficient per sq. ft. per 1 F difference is 0.45 as compared to 1.13 for single sash. A 1½-in. steam line rests on the sill between each sash and is supplied with steam at 2 lb. pressure. This arrangement is to provide heat on the inner glass surface to prevent condensation there during cold weather.

Since the concrete floor slabs extend to the outside in this type of building, it was found that in weather below 20 F the surfaces of the floors next to the outer walls became cold enough to cause a deposit of moisture on them. After some thought, this condition was corrected by installing a ¾-in. steam line, provided with steam at 2 lb. pressure, placed at the junction of the floors with the outside walls. This pipe was covered with a curved metal shield, with a concrete curb built over it. This extends into the room only 4 in. and does not subtract from the usable floor space. Valves were provided at suitable locations to enable the operators to control the colder zones as needed.

Processing Rooms

All processing is carried on in rooms especially designed and insulated for the particular purpose.

Four drying rooms, of 23,000 cu. ft. each, operating at approximately 95 F were constructed of 6-in. tile walls with 2-in. corkboard applied in ½-in. portland cement plaster and with two coats of the same finish.

Five fermentation rooms of approximately 13,000 cu. ft. each, with operating temperatures ranging up to 125 F, are constructed in the same manner on the sides, but with 2-in. cork applied to the floor slabs above and below the rooms, giving a completely insulated enclosure.

Five process rooms of approximately 9,000 cu. ft. with temperatures ranging from 62-66 F and relative humidities from 90 per cent to 95 per cent are in use. On some sides of these rooms, where space was restricted and maximum insulation desired, 3-in. corkboard was erected between 3-in. studs, with metal lath on one side. Two coats of plaster were then applied to each side. In other cases the usual tile wall was constructed and 3-in. corkboard applied against it, with finish coats on both tile and cork.

The Advantages of Insulating Industrial Plants

With a structure of this type, the following advantages have made themselves apparent:

Sketch of the Main Office and Plant of Bayuk Cigars, Inc., at Philadelphia. Open Windows Are Used in Seventh Floor Offices and Tenth Floor Spray Pond Only; All Others Remain Closed

1. Because of the low heat loss through the walls and double sash, it was not necessary to install direct radiation around the walls or under the windows. All heating is done by extended surface radiators in the ducts.

2. Elimination of objectionable condensation.

The minimum dew point of 65 F which must be maintained during the entire year necessarily means that the inner surfaces of the building must be maintained above this temperature. This is accomplished in the Bayuk building, and the streams of water which are so often seen trickling down the walls and windows of highly humidified industrial buildings during cold weather are absent. The health of operatives stationed near the outer walls is not endangered by dampness, and damage to materials and equipment is avoided.

3. Dehumidification is made economical.

This plant is equipped with a 100 ton centrifugal refrigeration unit used in conjunction with the air conditioning systems whenever the outside wet bulb temperature exceeds the 70 F wet bulb temperature desired in the building. This feature of the air conditioning system is of great value in preventing regain of prepared tobaccos under the humid summer conditions. The comfort provided for the operators assists materially in maintaining production schedules throughout the year.

Computations Show Savings: From the computation below, it is evident that more than 75 tons additional refrigeration would be required had the double sash and insulation not been used. This is based on a maximum differential of 20 F between the inside and outside.

			Heat Transmission			
		Temperature Difference	Without Insulation and with Single Sash		With Insulation and with Double Sash	
Surface	Sq. Ft.		Coeff.	Btu	Coeff.	Btu
Walls	87,000	20 F	0.295	513,300	0.097	168,780
Roof	25,200	20 F	0.500	252,000	0.115	57,960
Windows	27,000	20 F	1.13	610,200	0.45	243,000
				1,375,500		469,740

Subtracting the transmission from the higher figure, we find that 905,760 Btu per hour are eliminated from the cooling load. This is equivalent to 75.4 tons. These special features of construction effect a still further reduction of refrigeration requirements through the materially lowered rate of infiltration which is obtained.

4. Close control of humidity and temperature is possible.

Wide variations in the outside temperature, humidity, and wind velocity have been found to have no effect on the inner atmosphere. The air conditioning systems are able to operate at their maximum efficiency and the

View of Double Sash with One Pane of Glass Removed to Show the Heating Pipe Which Prevents Condensation on the Inner Glass Surface During Cold Weather

small permissible variation of only 2 per cent in relative humidity is not exceeded over any area.

5. The high temperature processing rooms operate without causing condensation to take place on the inner surfaces. Here, too, moisture dripping from the ceiling often results in damaged tobacco and equipment when these high dew points are used in ordinary rooms.

In the case of the low temperature rooms the outside of the walls is free of condensation.

All processes operate efficiently and without undue demands on steam and refrigeration. The areas adjacent to these rooms, particularly in the case of those of the higher temperatures, can be used for any type of manufacturing because of the low heat transmissions from the insulated walls of the process rooms. This also relieves the air conditioning systems of undue load in these work room areas.

6. Savings in equipment and operating costs.

The following table shows the estimated saving in initial boiler equipment due to the double sash and insulation, based on heating the building from 0 F to 70 F:

			Heat Transmission			
		Temperature	Without Insulation and with Single Sash		With Insulation and with Double Sash	
Surface	Sq. Ft.	Difference	Coeff.	Btu	Coeff.	Btu
Walls	87,000	70 F	0.295	1,796,550	0.097	590,730
Roof	25,200	70 F	0.500	882,000	0.115	202,860
Windows	27,000	70 F	1.13	2,135,700	0.45	850,500
				4,814,250		1,644,090

Again deducting, we find that 3,170,160 Btu per hour would be required to make up the difference. Dividing this by 33,479 it appears that 94 boiler hp would be required to supply the additional heat lost by transmission through uninsulated walls and single sash. Since this Bayuk plant has an installed boiler capacity of 185 hp for heating purposes, the saving effected here equals over 50 per cent. This percentage is also increased by reason of the reduced infiltration.

Since proper conditions in an efficient cigar plant must be maintained throughout the 24 hours of the day, and seven days in the week, the saving in fuel during the heating season may be computed as follows. An average outside temperature of 42 F is used here.

		Temperature	Heat Transmission			
			Without Insulation		With Insulation	
Surface	Sq. Ft.	Difference 70—42 F	Coeff.	Total Btu Per Hr.	Coeff.	Total Btu Per Hr.
Walls	87,000	28 F	0.295	718,620	0.097	236,292
Roof	25,200	28 F	0.500	352,800	0.115	81,144
Windows	27,000	28 F	1.13	854,280	0.45	340,200
				1,925,700		657,636

Deducting the present loss from the higher figure, we find that 1,268,064 Btu can be used as the average loss per hour during the entire season in an unprotected structure.

This is equivalent to 37.8 boiler hp per hour and multiplying by 24 equals 907.2 hp-h per day. Since operating tests have shown a consumption of one gallon of oil for every 2.7 boiler horsepower hours developed, the economy in fuel per day amounts to 336 gallons.

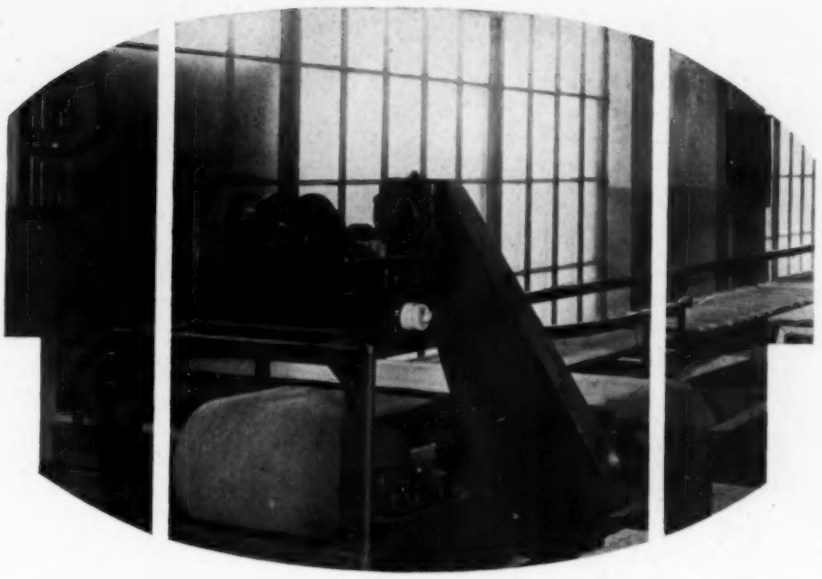
Based on a total number of 210 days in the season, the saving for the year equals 70,560 gallons.

With the additional savings due to reduced infiltration and lowered labor and depreciation charges, the total yearly saving represents a considerable return on the cost of the insulation and extra sash.

7. While shut-downs of the air conditioning equipment have been infrequent and of short duration, the conditions in the building have been found to maintain themselves surprisingly well during such periods and a minimum length of time is required to return them to normal.

Primary Object to Improve Product

The primary purpose of the Bayuk management using this insulation and other unusual features of construction



Conveyor Delivering Stripped Tobacco to Drying Rooms on Next Floor. This Could Only Be Possible in an Insulated, Conditioned Building

was to provide the most favorable conditions for the treatment of delicate tobaccos and the manufacture of fine cigars. This object has been achieved, for all classes and grades of leaf tobacco are maintained at the correct moisture content, in every stage and process. As a natural result, the finished cigar is provided with that quality so essential to its enjoyment—perfect smoking condition.

Avoiding Shock When Leaving Air Conditioned Spaces

In air conditioning buildings the contrast between outside and inside conditions is a major factor in deciding what temperatures and humidities should be maintained. Consideration must be given to two classes of people; those who enter the conditioned area for only a short time, and those who spend the entire day in the conditioned space. If the contrast is too great, the first class is subjected to two distinct changes; the first, on entering the area, may not be an unpleasant one except in extreme cases, where the effect is that of walking into a refrigerator; but the second, upon leaving, is a severe one. This latter is generally the result of too great a temperature difference accompanied by a saturation of the clothing, effected by a high moisture content in the conditioned area. This combination of moist clothing and sudden contact with high street temperature finds the body with no means of throwing off the heat load and the effect is quite noticeable.

These considerations point to the desirability of an inside dry bulb temperature differing from that of the outside air in the order of about 15 degrees under maximum outside conditions and approaching more nearly to the outside temperature as the latter approaches 80 F. If, in addition to a temperature differential as indicated above, the proper relative humidity is maintained, sufficient to insure rapid removal of body moisture, the conditions within the area are extremely comfortable. Not only is the comfort of the transient population of the conditioned area catered to by reducing the entrance and exit shocks but further, the resident population is cared for by providing the humidity potential necessary to keep the skin and clothing free of moisture and at a temperature insuring complete comfort.

The importance of this "dryness factor" cannot be too greatly emphasized. Actual observations have shown that a feeling of invigoration is sometimes noticed when one passes into an area where the dry bulb temperature is somewhat higher but the relative humidity is considerably lower than that just left. This is only true within certain limits, of course, but it is felt that the major part of discomfort attendant to high temperature is caused by accompanying high relative humidity. A consideration of the foregoing would seem to point to an established maximum of 85 F dry bulb in the populated zone where the outside temperature is in the order of 100 F. A relative humidity of 40 per cent at 85 F will prove comfortable under such conditions.—*Eugene D. Milener,* before the natural gas department of the *American Gas Association* at Memphis.

HART HALL, A DORMITORY AT TEXAS A. & M. COLLEGE

By
F. E. Giesecke*

How to Design the Piping

for a Combined Forced and Gravity Circulation Reversed Return

Hot Water Heating System

By balancing the friction losses through the various circuits of a hot water heating system, the proper amount of hot water will be supplied to each radiator and the success of the installation assured. Frequently, orifices are used to assist in this procedure; the selection of the proper pipe sizes is, of course, of paramount importance.

Professor Giesecke needs no introduction as an expert in designing hot water heating systems. In this article he explains the design of the piping for the heating system in a college dormitory of four floors. As the method is explained in detail, it may be applied with success to your own jobs.

THE building to be heated, Hart Hall, is a dormitory for cadets at the Texas A. & M. College and has 148 rooms.

The heating system is a combined forced and gravity circulation reversed return water system with supply and return mains located underneath the first floor and with 39 sets of flow and return risers.

The mains are located so that they form two separate and distinct circuits, one having a length of about 340 feet and 17 sets of risers and shown in Fig. 2, and the other having a length of about 390 feet and 22 sets of risers. The two circuits are connected to the main flow and return line near the center of the building.

The two circuits were designed as described below for the shorter one, shown in Fig. 2, which supplies the water to 17 sets of risers and to 67 radiators.

The pressure head which causes the water to flow through the several radiators is produced partly by a circulating pump located in the power house, about 3,600 feet from the building, and partly by the difference in the weight of the water in the flow and return risers.

Identifying the Radiators

The risers are numbered consecutively from *1* to *17* and the radiators served by the risers are designated by the corresponding riser numbers with the addition of the letters *a, b, c,* and *d,* representing, respectively, the first, second, third, and fourth floors; thus, Radiator *15c* is the third floor radiator served by Riser *15.*

The Problem in Design

A reversed return system of piping is a two-pipe system; i. e., a system in which the water flows through one radiator only before it returns to the central plant to be reheated.

The circuit under consideration supplies water to 67 radiators and hence the water which enters the circuit may follow any one of 67 paths. The problem is

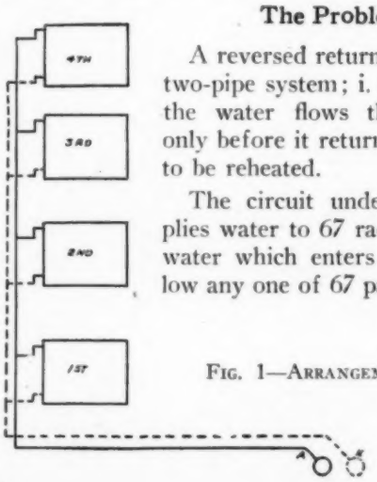


FIG. 1—ARRANGEMENT OF RADIATORS

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to design this system so that the water will distribute itself among the 67 paths so that each of the 67 radiators will receive its correct share of hot water. Considerable latitude exists for the selection of the pipe sizes for the mains and for the risers. If the selected pipe sizes are comparatively large, the power required to force the water through the circuit will be comparatively small. There is an optimum pipe size with a corresponding power requirement which will constitute the most economical installation. This particular design can be determined by making several trials and then selecting the one which is most economical.

The Procedure

For the case in hand, it was assumed that an average velocity of about two feet per second or a total friction head of about three feet of water in the mains of each of the two circuits would result in an economical installation.

Since the length of the short circuit is about 340 feet, its pipe sizes must be such that the average friction head is about 36000/340 or about 100 milinches per foot of pipe. (Note. Three feet of water is equal to 36000 milinches.)

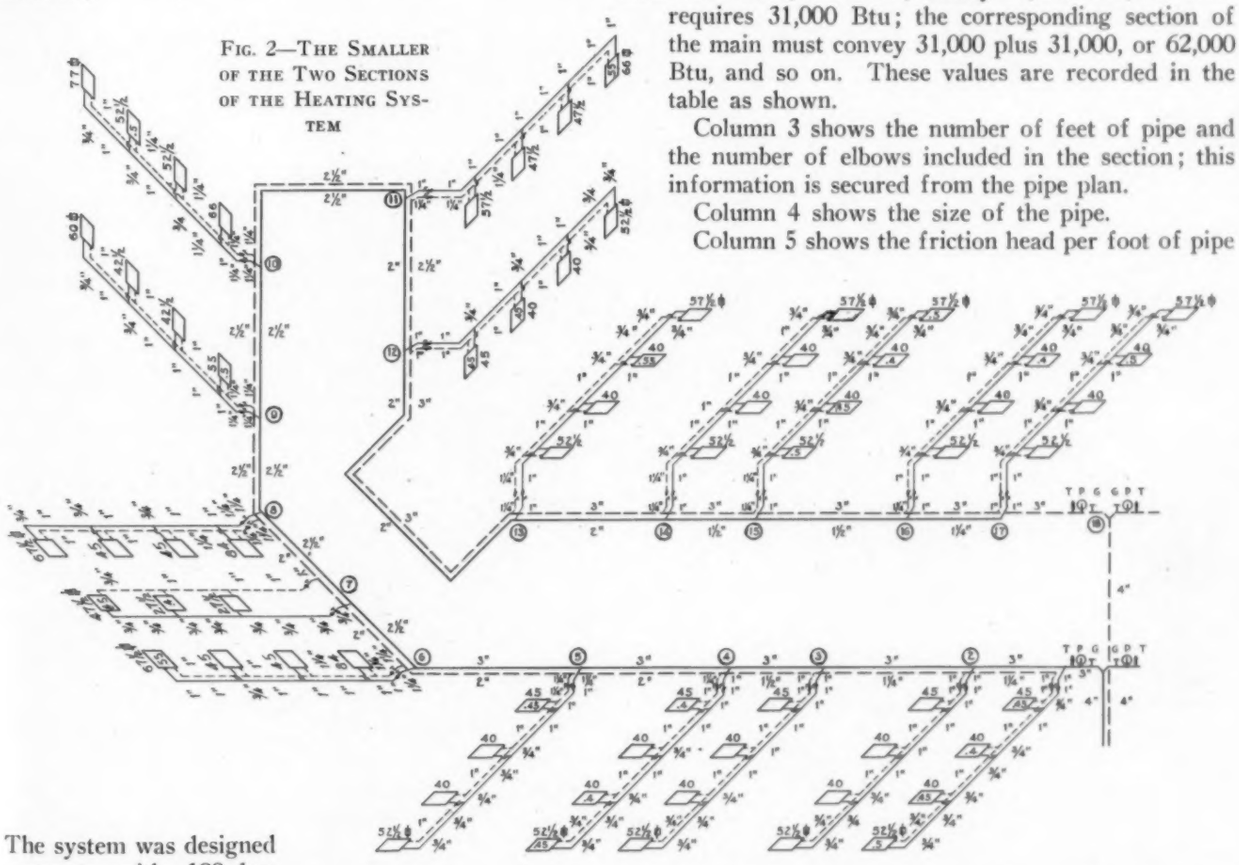


FIG. 2—THE SMALLER OF THE TWO SECTIONS OF THE HEATING SYSTEM

The system was designed to operate with 190-deg. water in the flow lines and 170-deg. water in the return lines when the outside temperature is 25 degrees and the inside temperature 70 degrees.

From a pipe chart, it appears that for a friction head of about 100 milinches per foot of pipe and a temperature drop of 20 degrees the following quantities of heat will be conveyed by the respective pipes:

Three in., 550,000 Btu; 2½ in., 300,000 Btu; 2 in., 190,000 Btu; 1½ in., 100,000 Btu; 1¼ in., 62,000 Btu.

With these data a tentative selection of the pipe sizes for the flow and return mains was made and recorded in Table 1, as shown.

Table 1 Explained

Table 1 was prepared as follows: Column 1 shows the designations of the sections into which the flow main is divided. A section is a subdivision of the main extending either from the beginning of the main to the first riser connection, or from the first riser connection to the second and so on. Every section of the flow main has a definite quantity of water flowing through it and this quantity generally differs from that flowing through any other section of the same circuit.

Column 2 shows the quantity of heat conveyed by each section of the flow main; this information is secured by beginning with the last riser connection and adding, successively, the quantities of heat required by the several sets of risers. The quantities are derived from the calculations of the heat losses of the several rooms served by the respective risers and are recorded on the plan near the risers. For instance, the last riser, Riser *17*, requires 31,000 Btu; the last section of the flow main must, therefore, convey 31,000 Btu; Riser *16* requires 31,000 Btu; the corresponding section of the main must convey 31,000 plus 31,000, or 62,000 Btu, and so on. These values are recorded in the table as shown.

Column 3 shows the number of feet of pipe and the number of elbows included in the section; this information is secured from the pipe plan.

Column 4 shows the size of the pipe.

Column 5 shows the friction head per foot of pipe and elbows. This information is secured from the pipe chart in the usual way.

Column 6 shows the friction head for the whole section.

Column 7 shows the progressive sums of the friction heads in the successive sections to the several riser connections. Thus, for the first riser connection, the friction head is 465 milinches; for the second riser connection, it is 465 plus 1,105, or 1,570 milinches, and so on

TABLE 1—FRICTION HEADS IN FLOW AND RETURN MAINS

Flow Main							Return Main								
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
Section	1000 Btu	Feet Ells	Diameter	Unit Frict.	Total Frict.	Grand Total	Section	1000 Btu	Feet Ells	Diameter	Unit Frict.	Total Frict.	Grand Total	Final Total	Risers
0–1	533.7	5	3	93	465	465	1–2	28.7	13	1¼	23	299	29082	29547	1
1–2	505.0	13	3	85	1105	1570	2–3	57.4	21½	1¼	80	1720	28783	30353	2
2–3	476.3	21½	3	76	1634	3204	3–4	86.1	13	1½	78	1014	27063	30267	3
3–4	447.6	13	3	68	884	4088	4–5	114.8	21½	2	40	860	26049	30137	4
4–5	418.9	21½	3	60	1290	5378	5–6	143.5	23½	2	60	.....	.....	.....	.....
5–6	390.2	23½	3	53	.....	.....	.....	.....	1–45°	2	190	1600	25189	30567	5
.....	.....	1–45°	3	260	1505	6883	6–7	183.5	18	2	95	1710	23589	30472	6
6–7	350.2	13	2½	135	1755	8638	7–8	200.3	12	2	110	1320	21879	30517	7
7–8	333.4	18½	2½	120	2220	10858	8–9	240.3	15½	2½	65	.....	.....	.....	.....
8–9	293.4	14	2½	95	.....	.....	.....	.....	1–45°	2½	260	1268	20559	31417	8
.....	.....	1–45°	2½	360	1690	12548	9–10	273.5	21	2½	82	1722	19291	31839	9
9–10	260.2	21	2½	75	1575	14123	10–11	314.0	31½	2½	110	.....	.....	.....	.....
10–11	219.7	31½	2½	55	.....	.....	.....	.....	2–90°	2½	600	4665	17569	31692	10
.....	.....	2–90°	2½	320	2373	16496	11–12	350.0	21½	2½	135	2903	12904	29400	11
11–12	183.7	21½	2	100	2150	18646	12–13	378.7	50	3	50	.....	.....	.....	.....
12–13	155.0	54½	2	70	.....	.....	.....	.....	2–45°	3	245	.....	.....	.....	.....
.....	.....	2–45°	2	217	.....	.....	.....	.....	2–90°	3	350	3690	10001	28647	12
.....	.....	2–90°	2	310	4769	23415	13–14	409.7	21½	3	58	1247	6311	29726	13
13–14	124.0	21½	2	50	1075	24490	14–15	440.7	13	3	67	871	5064	29554	14
14–15	93.0	13	1½	92	1196	25686	15–16	471.7	21½	3	75	1613	4193	29879	15
15–16	62.0	21½	1½	45	968	26654	16–17	502.7	13	3	84	1092	2580	29234	16
16–17	31.0	13	1¼	27	351	27005	17–18	533.7	16	3	93	1488	1488	28493	17

to 27,005 milinches for the last or 17th riser connection.

Columns 8 to 14 show the corresponding values for the return main. However, it should be noted that the totals in column 14 are secured by beginning with 1,488 milinches, the friction head for the last or 17th riser connection, and adding, successively, to 29,082 milinches, which is the friction head in the return main for the first riser connection.

Column 15 shows the sum of the total friction heads in the flow main for any one riser connection and the total friction heads in the return main for the same riser connection. Thus, for Riser Connection No. *1*, the total friction head is equal to the friction head in the first section of the flow main (465 milinches) plus the friction head in all sections of the return main (29,082) or to 29,547 milinches, as shown in Column 15.

Similarly, for Riser Connection No. *10*, the friction head is equal to 14,123 from Column 7 plus 17,569 from Column 14, or 31,692 milinches, as shown in Column 15.

In the same manner, the figures shown in Column 15 give the total friction head, partly in the flow main and partly in the return main, which must be overcome in order to make the proper quantity of water flow to that particular riser.

For an ideal arrangement of the pipe sizes in the flow and return mains, the values shown in Column 15 should all be the same. This condition cannot, as a rule, be attained with standard pipe sizes available.

A study of the seventeen values shown in Column 15 shows that the lowest and highest values are 28,493 and 31,839 milinches. The difference between these two values is 3,346, or about ten per cent of the highest value. This difference is probably small enough so that the pipe sizes shown in Table 1 may be adopted for the final design. If the difference were much larger it might be better to vary the pipe sizes so as to secure more nearly uniform results. In fact, the sizes shown in Table 1 are not those which were first selected, but those which were selected after a few trials and substitutions had been made. Having thus completed the design of the flow and return mains, the next step is to design the riser connections to and through the first floor radiators.

Riser Connections to First Floor Radiators

For these connections we may arbitrarily assume that no pipe shall be larger than 1¼ inches and none smaller than 1 inch, except the radiator connection itself, which may be ¾ inch, and that there shall be no change in size in the section between the connection to the main and the connection to the radiator except at the union in that section. This would make possible five major combinations of 1¼ inch and 1 inch pipe. In order to reduce the differences which exist in the values shown in Column 15, it will be best to select the 1 inch pipe for the flow and return connections for those risers which have the low values in Column 15; the 1¼ inch pipe for those which have the high values in Column 15, and one of the several possible combinations of the 1 inch and 1¼ inch pipe for those which have the intermediate values.

Beginning then with Riser Connection No. *9*, which has the largest friction head, 31,839 milinches, let Fig. 3 represent that connection, divided into five sections, *AB*, *BC*, *CD*, *DE*, and *EF*. The data relating to this section are shown in the usual way in Table 2. They were derived by selecting the largest pipe sizes proposed, namely, 1¼ inches for the two risers and 1 inch for the radiator connections.

The friction head for these five sections is 1,414 mil-

inches, which, added to 31,839, gives a total of 33,253 milinches for the circuit of this particular radiator, *9a*.

Proceeding next with Riser Connection No. *17*, which has the smallest friction head, 28,493 milinches, and assigning to it the smallest pipe sizes proposed, namely, 1 inch for the risers and ¾ inch for the radiator connections, the corresponding friction is, as shown in Table 3, 5,038 milinches. Adding this to 28,493 gives 33,531 milinches for the total through the Radiator *17a*.

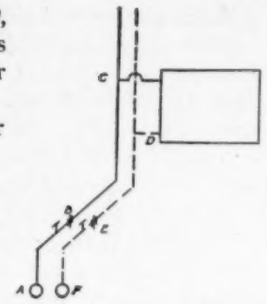


FIG. 3—CONNECTION OF MAIN TO FLOW AND RETURN RISERS NUMBER 9

The difference between the friction heads of Radiator *9a* and Radiator *17a* is 33,531 — 33,253, or 278 milinches, less than 1 per cent of the total friction head. This difference is small enough to be neglected.

The next step is to design the connections of the remaining 15 sets of risers so that, in every case, the friction head through the first floor radiator is as nearly as possible equal to 33,531 milinches. To do this a table is prepared, similar to Tables 2 and 3, for each of the 15 sets of risers and the pipe sizes for the five sections, *AB, BC, CD, DE,* and *EF* selected so that the total friction head is as near 33,531 as possible.

This was done. The pipe sizes selected are shown on the pipe diagram of Fig. 2. The velocity of the water in the respective radiator connections and the total friction heads for the seventeen sets of risers are recorded in Table 4. When the friction head exceeds 33,531, the deficit is recorded as negative.

The friction heads in the seventeen circuits shown in this table are quite uniform. The difference between the highest and lowest is 710 milinches, or 2.1 per cent of the highest, or 2.15 per cent of the mean of the seventeen values.

TABLE 4—TABULATION OF THE FRICTION HEADS THROUGH THE SIXTEEN FIRST FLOOR RADIATORS

Riser Connection	1	2	3	4	5	6	8	9	10
Total Frict.	33086	33614	33528	33058	33210	33711	33645	33253	33768
Deficit	445	–83	3	473	321	–180	–114	278	–237
Radiator Connec.	¾	1	1	¾	1	1	1	1	1
Velocity	5½	3	3	5½	3	6	6	4	5
Orifice	.45			.45	0.5				

TABLE 4—(Continued)

Riser Connection	11	12	13	14	15	16	17
Total Frict.	33742	33079	33238	33520	33390	33567	33531
Deficit	–211	452	293	11	141	–36	0
Radiator Connec.	1	¾	¾	1	¾	¾	¾
Velocity	4	5½	6	4	6	6	6
Orifice		.45					

Use of Orifices

This difference is probably much less than the difference between the assumed outside temperature and the actual outside temperature and probably also much less than the difference between the calculated heat loss of one of the rooms and the actual heat loss of that room for a given outside temperature, and the pipe system as designed will give entirely satisfactory results. If, however, the differences between the several friction heads are to be reduced still further, additional friction may be introduced by inserting diaphragm orifices in unions placed in the radiator connections (or in any other portion of the circuit).

To determine the rates at which such diaphragm orifices cause friction heads, investigations were conducted by the *Texas Engineering Experiment Station* in cooperation with the *American Society of Heating and Ventilating Engineers*. The results are shown by the curves of Figs. 4 and 5. To explain the use of these curves, note, from Table 4, that the circuit of Radiator *1a* should have 445 milinches additional friction; note also that the velocity of the water in the ¾ inch radiator is 5½ inches per second. Refer to Fig. 4 and note that

TABLES 2 AND 3—TOTAL FRICTION HEADS THROUGH TWO OF THE FIRST FLOOR RADIATORS

Radiator 9a

Section	1000 Btu	Feet Elbows	Size	Velocity In.	Unit Friction	Total Friction
AB	33.2	2 34	1¼		32 80	336
BC	33.2	5 1.5	1¼		32 80	280
CD	9	2 11.2	1	4	11 20	246
DE	33.2	3 1.5	1¼		32 80	216
EF	32.2	2 3.4	1¼		32 80	336
Total						1414
Friction in Main						31839
GRAND TOTAL						33253

Radiator 17a

Section	1000 Btu	Feet Elbows	Size	Velocity In.	Unit Friction	Total Friction
AB	31	2 3.4	1		110 230	1000
BC	31	9 1.5	1		110 230	1335
CD	8.5	2 11.2	¾	6	31 47	588
DE	31	7 1.5	1		110 230	1115
EF	31	2 3.4	1		110 230	1000
Total						5038
Friction in Main						28493
GRAND TOTAL						33531

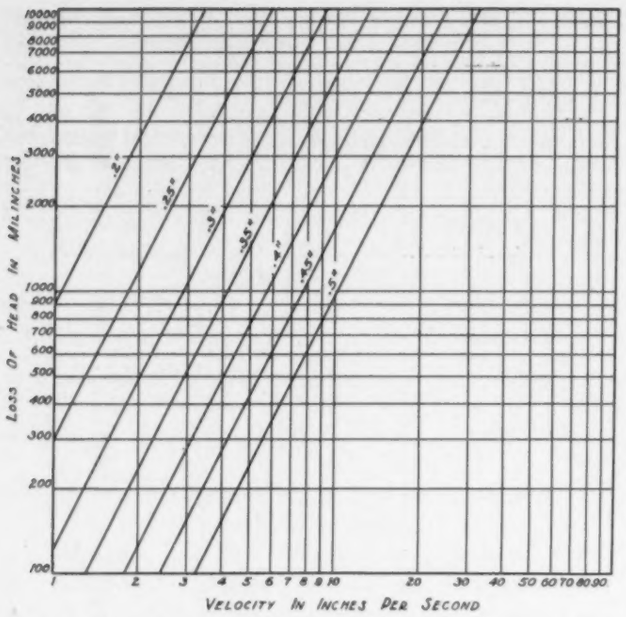


FIG. 4—FRICTION HEAD OF CENTRAL CIRCULAR DIAPHRAGM ORIFICES HELD BY UNIONS IN A ¾-IN. PIPE LINE

for a velocity of 5½ inches and a friction head of 445, a 0.5 inch orifice should be used.

Note also that for Radiator *5a,* 321 milinches additional friction should be supplied. The velocity in the 1 inch radiator connection is 3 inches per second. From Fig. 5, a 0.5 orifice should be used. In this manner the orifices were selected which are shown in Table 4, and in Fig. 2.

For those cases in which the necessary additional friction head is less than 1 per cent of the total friction head, i. e., in this case less than 335 milinches, no orifice was provided except for Radiator *5a,* discussed above to illustrate the use of the curves of Fig. 5.

Allowable Tolerance

In order to form an opinion as to the tolerance which may be permitted in a series of radiator circuits in a forced circulation system, without interfering appreciably with the satisfactory operation of the system, let us examine Fig. 6, which shows a small system.

One radiator is to dissipate 15,000 Btu, the other 10,000 Btu. The pipe sizes were selected so that the main is 1¼ inches and the branches supplying the two radiators are, respectively, 1 inch and ¾ inch. Each branch is to be composed of 15 feet of pipe and of 14 elbow equivalents.

The temperatures of the water in the flow and return lines are to be 190 and 180 degrees. The sizes of the two radiators, for $K = 1.4$ when the temperature difference is 115, should be 93 and 62 square feet.

By reference to a pipe diagram we find that the friction heads in the 1 inch and in the ¾ inch circuits will be, respectively, 4,580 and 6,320 milinches, if the system functions as specified. The difference between these two heads is 1,740 milinches, or 27 per cent of the larger value. It is evident that this condition cannot exist because the friction heads in the two branch circuits, connecting the 1¼ inch pipe at the points *A* and *B,* must be equal to each other.

A calculation will show that each of the two friction heads will be about 5,350 when the water is circulating through the system at such a rate that 25,000 Btu will be delivered to the two radiators when the temperature drop in the system is 10 degrees. The two calculated or desired friction heads differ from the actual friction head by 14 per cent and 18 per cent, respectively.

In order to increase the friction head in the circuit of the large radiator from 4,580 to 5,350 it is necessary that the water have a velocity 16/15 of that implied when the 1 inch pipe was selected for that circuit. The water will then remain in the large radiator only 15/16 of the time implied in the design and the temperature drop through the radiator will be only 15/16 of 10 degrees, or 9.4 degrees. The average water temperature in the radiator will be 186.3 degrees and the heat dissipated 15,150 Btu instead of 15,000, as calculated, or 1 per cent more.

Similarly, in order to decrease the friction head in the circuit of the small radiator from 6,320 to 5,350 it is necessary that the water have a velocity 9/10 of that implied when the ¾ inch pipe was selected for that circuit. The water will remain in the small radiator 10/9 the time implied in the design and the temperature drop through the radiator will be 10/9 of 10 degrees, or 11.1 degrees. The average water temperature in the radiator will be 184.5 and the heat dissipated 9,940 Btu instead of 10,000 Btu as calculated, or 6/10 of one per cent less.

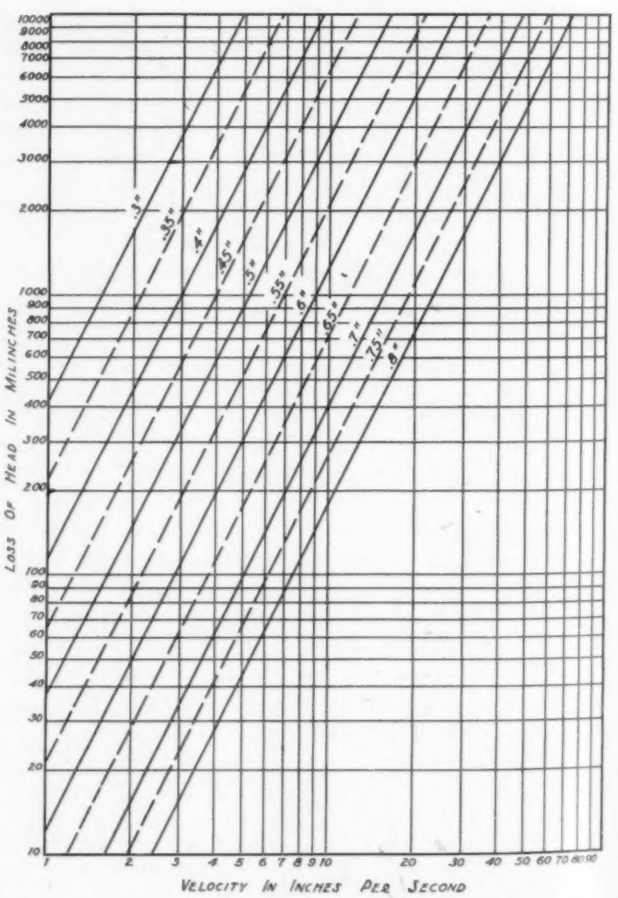


FIG. 5—FRICTION HEAD OF CENTRAL CIRCULAR DIAPHRAGM ORIFICES HELD BY UNIONS IN A 1-IN. PIPE LINE

It is evident, from this discussion, that an inaccuracy in the pipe design of a small forced circulation system like that shown in Fig. 3, which results in an error of about 15 per cent of the friction head, causes an error

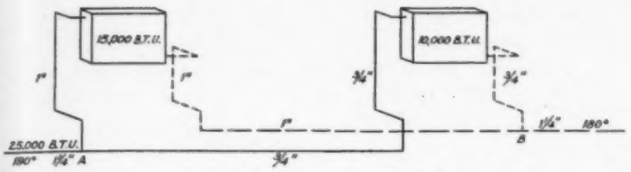


FIG. 6—A SMALL HEATING SYSTEM WITH FORCED CIRCULATION TO EXPLAIN THE EFFECT OF ERRORS IN THE SELECTION OF PIPE SIZES

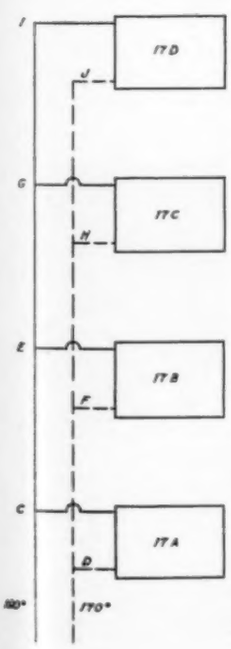
of only about 1 per cent in the heat dissipated by the radiators.

For complicated systems, like that shown in Fig. 2, which has 34 times as many circuits as that shown in Fig. 3, it would be quite difficult to calculate the effect of an error in the design of the pipe system upon the heat dissipation of the several radiators, but, from the above calculation for a simple system, it seems perfectly safe to conclude that if the pipe system is calculated so accurately that the calculated friction heads do not differ more than 1 per cent from those which will actually obtain in the system, the operation of the system will be entirely satisfactory and it is usually not necessary to secure greater accuracy by the introduction of diaphragm orifices. However, this statement applies only to forced circulation systems. For gravity circulation systems additional factors must be considered and the problems involved therein are more complicated.

Balancing the Remaining Circuits

Having balanced the first floor radiators of the seventeen sets of risers so that the friction head of every circuit is practically 33½ inches, the friction heads of the remaining radiator circuits must be adjusted in a similar manner.

FIG. 7—FLOW AND RETURN RISERS NUMBER 17



Beginning with Riser *17* shown in Fig. 7, when the water has entered the flow riser and reached the point *C* it may flow through the first floor radiator and enter the return riser at *D*, or it may continue in this flow riser to the points *E*, *G*, or *I*, and flow through the second, third, or fourth floor radiators and enter the return riser at *F*, *H*, or *J*, respectively.

If the system is designed correctly, each of the four radiators will receive its proper quantity of hot water. This condition is attained when the friction head in each of the four possible paths (which the water may follow in flowing from *C* to *D*) is exactly equal to the pressure head available for those four paths.

From Table 3 we see that the friction head in the circuit of Radiator *17a* is 33,531 milinches and that 588 thereof is in Section *CD* through the radiator. When the system is functioning correctly there is available, at the point *C*, a pressure head of 588 milinches tending to force the water through Radiator *17a*. This same pressure head of 588 milinches tends also to force the water through Radiators *17b*, *17c*, and *17d*. In addition to this pressure head of 588 milinches, there are available, for the three upper radiators, three gravity heads corresponding to their elevations above the first floor radiators. These elevations are, respectively, 9.25, 18.50, and 27.75 feet. The corresponding pressure heads are, for temperatures of 190 and 170 degrees, 796, 1,592, and 2,388 milinches, and the total available pressure

TABLE 5—FRICTION HEADS THROUGH THE THREE UPPER FLOOR RADIATORS FOR RISERS NUMBER SEVENTEEN

Radiator 17b, P. H. 1384

Section	1000 Btu	Feet Elbows	Size	Velocity In.	Unit Friction	Total Friction
CG	22.5	9.25	1		58	539
GH	6.5	2 11.2	¾	4.5	19 27	341
HD	22.5	9.25	1		58	539

Total	1419
Deficit	—35
Orifice	None

Radiator 17c, P. H. 2180

CG						539
GI	16	9.25	1		30	279
IJ	6.5	2 11.2	¾	4.5	19 27	341
JH	16	9.25	1		30	279
HD						539

Total	1977
Deficit	203
Orifice	.5

Radiator 17d, P. H. 2976

CG						539
GI						279
IK	9.5	9.25	¾		36	335
KL	9.5	2 10	¾	7	36 56	632
LJ	9.5	9.25	¾		36	335
JH						279
HD						539

Total	2938
Deficit	38
Orifice	None

heads for the three upper radiators are 1,384, 2,180, and 2,976 milinches.

The pipe sizes for the three circuits are determined in the usual way as indicated in Table 5. Only one orifice is needed for this group.

The other sixteen groups of radiators were designed in a similar manner. The orifices selected are shown on the respective radiators in Fig. 2.

Riser 7 has no first floor radiator. It must evidently be designed so that the friction head of the second floor radiator is, as nearly as practicable, equal to the friction head for the first floor radiators, 33,531 milinches, plus the excess of gravity head of a second floor radiator over that of a first floor radiator, 796 milinches, or 34,327 milinches.

The usual provisions must be made to provide for expansion, for venting, and for draining.

This completes the design of the system.

System Checked Last Winter

The heating system described above has been in continuous and satisfactory operation during the winter of 1930-31.

In order to secure approximate information regarding the relative heat emission by the radiators on the four floors of the building, it was assumed that the relative number of open windows on the four floors would be an index of the relative state of the heating of those floors.

The open windows were counted twenty times during the colder part of the season with the following average results: First floor, 46; second floor, 42; third floor, 42; and fourth floor, 37 per cent of the total number of windows on the respective floor were partially open.

According to this record, the first floor seems to have been slightly overheated and the fourth floor slightly underheated when compared with the second and third floors. This difference may have been due to several causes, for example, to an inaccuracy in calculating the heat losses of the several floors, to an inaccuracy in calculating pipe sizes, or to temperamental differences in the students living on the several floors of the building.

It should be noticed that the radiators in the building are not provided with valves and therefore do not permit of temperature regulation in the individual rooms.

Two Supply Mains Used for Central Hot Water Heating

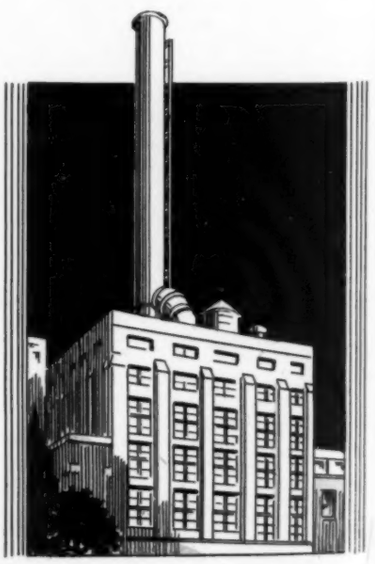
IN SWEDEN, hot water heating is used extensively. The radiators are frequently of welded steel. The convectors also usually are of steel, but convectors are not popular in that country because they are believed to cause more dust deposit on the walls than do exposed radiators.

A popular scheme for forced hot water circulation, especially for group heating, is to send the water out in two supply mains, in one of which the water is kept at 170 degrees for example, and in the other of which the water is kept at 150 degrees. They both serve a single return, in which the water is calculated, under the above stated conditions, to be at 140 degrees. At each building there are adjustable bypasses between the supply mains, and the buildings nearest the central station may thus be adjusted so as to receive their normal volume of water at the optimum temperature, anywhere between 150 degrees and 170 degrees without interference with the delivery of water, as hot as may be desirable, to the most remote building.

Special covered tool operated plug-valves are used for the minute and exact permanent adjustment of the volume of water going to each radiator.

Professor Gille, consulting engineer and university professor of Stockholm, states that welded steel sectional radiators as used in Sweden do not cause any difficulties due to corrosion, because with water in the interior corrosive action seems to be absent. This is at variance with experience in this country with steel push nipples between cast iron sections of radiators on hot water heating systems. The difficulty here may have been due to use of two different metals—whereas in Sweden but one metal is used.

Professor Gille, however, believes that if we use steam in the standard Swedish steel radiators we will have trouble, due to the chemical and electrical concomitants of the higher pressures and temperatures.—*S. R. Lewis.*



Piping Plant Atkinson

By E. C. Gaston*

A brief summary of the particular features of the Plant Atkinson piping as covered in this article:

The plant piping is divided into three classes: High pressure (600 lb. Std.), Intermediate pressure (250 lb. Std.), and Low pressure (125 lb. Std.).

Forged or cast steel valves and fittings, also special alloy steel bolts threaded the entire length with nuts on each end, are used on all intermediate pressure as well as high pressure lines.

Plain copper plated steel gaskets are used on all high pressure and intermediate pressure flanged joints, the flanges being faced with a fine phonographic finish with grooves not over 1-64 in. deep.

A piping gallery extends the entire length of the plant. It is centrally located and contains nothing but piping.

By-passes around the feedwater regulators were eliminated, resulting in a substantial saving due to the reduction in the number of high pressure valves and fittings required at these points.

Circulating water from the condenser is discharged to the river through a 72-inch underground cast iron bell and spigot pipe line. This is an outstanding feature, and was one of the principal factors contributing to the low unit cost of this plant.

Expansion of pipe lines has been adequately provided for through use of bends and properly located anchors, supports, and guides.

The thickness of heat insulating material was calculated for each pipe line. Special high temperature blocks are used only on main steam, and first extraction point lines. A special method is used to attach the insulation blocks to all gas and air ducts in order to provide a smooth surface over the entire area, including the stiffener angles.

The above features were all instrumental in producing a well designed job for a minimum first cost.

MANY articles have appeared recently describing the general features and particular problems incurred in the design and construction of Plant Atkinson, the new 60,000 kw steam plant of the Georgia Power Co., though the matter heretofore published passes over the station piping design and installation with little or no comment. This is probably as it should be, since the plant piping presents one of the major problems in the design of the station, and for this reason deserves more space than could be allotted to its discussion in describing the general plant features.

This article will be devoted entirely to the description of the actual piping and accessory equipment, and to the discussion of the major problems encountered in the design. In it, an effort will be made to describe the piping, the fittings, valves, gaskets, bolts, etc., giving reasons for their selection and the general specifications under which they were purchased.

The Principal Equipment

Plant Atkinson is located eight miles from Atlanta, Ga. The principal equipment consists of a 60,000 kw, single cylinder, condensing type turbo-generator with extraction outlets for four stage feedwater heating. Steam from the turbine is exhausted into a 70,000 sq. ft. two-pass condenser.

Steam for the unit is supplied from two 3,020 hp straight tube cross drum boilers, each being equipped with a 30,000 sq. ft. preheater, but without economizers. The furnaces are provided with water-cooled rear and side walls, and an air cooled front wall suspended above the burners. Ash hoppers are provided at the bottom of the furnaces for the removal of ashes by the jet sluicing system. The unit system of pulverized coal firing is employed.

Steam is produced at 475 lb. per sq. in. gage pressure and 725 F total temperature.

Importance and Variety of Piping

It can be well stated that a steam plant consists principally of a boiler, turbo-generator, a few pumps and heaters, and PIPING—not the least of which is piping. Not everyone realizes the importance of this item in modern steam plant practice, either from the quantities involved or from the problems in design and layout encountered. The following figures serve to give an idea of the quantities involved.

The total weight of all piping materials exclusive of valves, and special circulating water castings and discharge piping, amounts to 250 tons in Plant Atkinson. There are 30,000 lb. of bolts and nuts alone. The size of the pipe varies from ⅛-in. brass tubing, as used for instrument connections, to the 72-in. cast iron pipe used in the circulating water discharge line.

Piping May Be Controlling Factor in Design

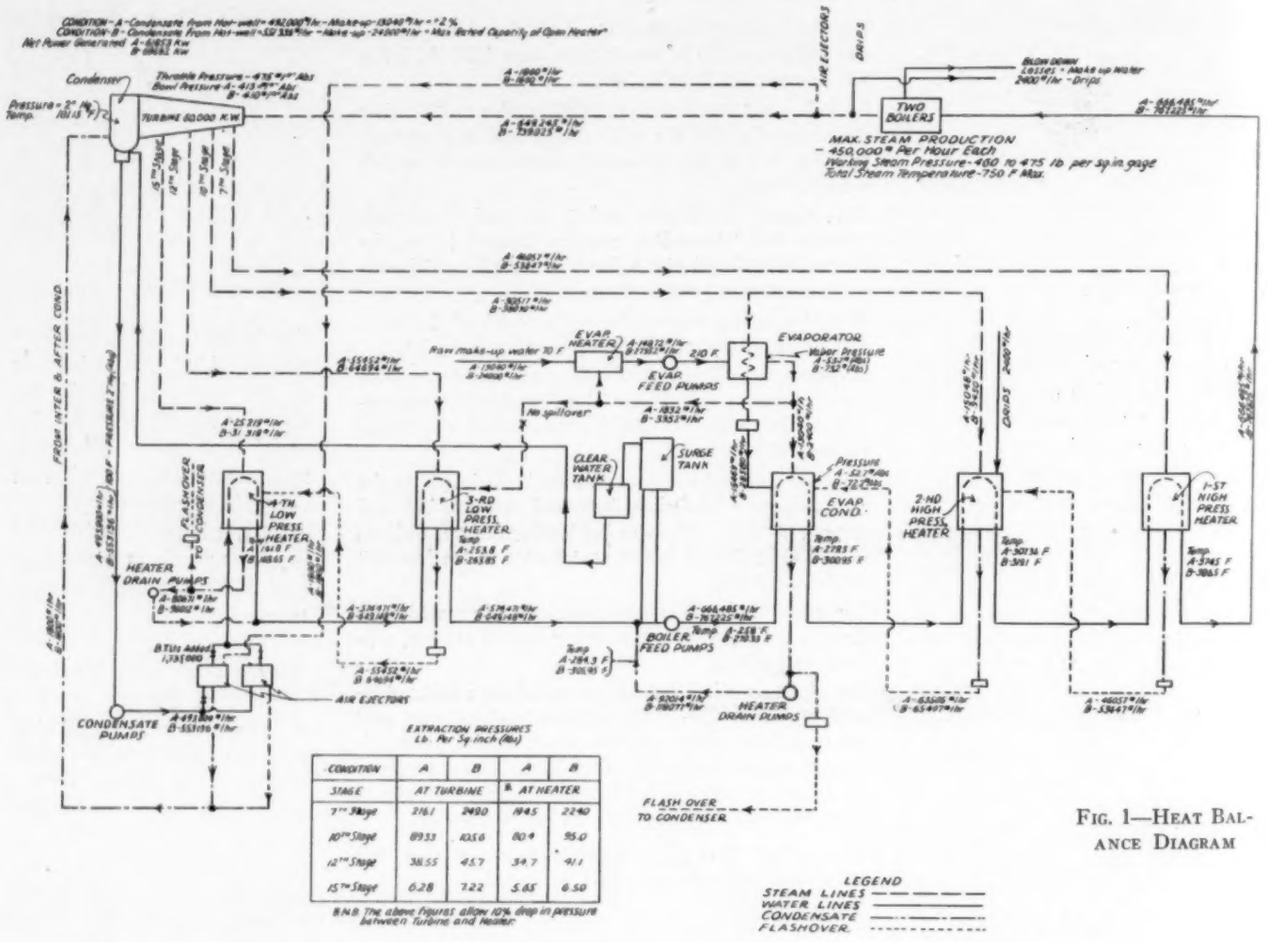
In order to insure a first class piping job it is essential that it be considered from the very beginning of the plant design. Many times the piping becomes the controlling factor in the location of the plant equipment. Improper location of equipment with regard to piping may result in improper design and layout of piping to care for expansion and pipe stresses, thereby producing an eternal source of worry and expense to the permanent plant operators through the medium of broken flanges, leaking

* Allied Engineers, Inc. (Southern Division), Birmingham, Ala.

joints, etc.; or in the attempt to provide the proper piping it may result in excessive cost by requiring special fittings, or excessive lengths of pipe in order to clear other equipment and provide for expansion.

The Three Classes of Piping

Piping in Plant Atkinson is divided into three classes: high pressure, intermediate pressure, and low pressure.



CONDITION	A	B	A	B
STAGE	AT TURBINE		# AT HEATER	
7th Stage	216.1	249.0	194.5	224.0
10th Stage	89.33	103.6	80.4	93.0
12th Stage	38.55	45.7	34.7	41.1
15th Stage	6.28	7.22	5.65	6.50

N.B. The above figures allow 10% drop in pressure between Turbine and Heater

FIG. 1—HEAT BALANCE DIAGRAM

High pressure piping includes the main steam, auxiliary steam, and feedwater lines, and was specified to be seamless tubing suitable for 600 lb. per sq. in. working pressure.

Intermediate pressure piping includes the extraction lines from the No. 1 and No. 2 bleed points to the feedwater heaters and evaporators, and was specified to be lap welded, open hearth steel pipe of standard thickness as specified for 250 lb. per sq. in. working pressure.

Low pressure piping includes all other steam, water, and air piping in the plant and was specified to be lap welded, open hearth steel pipe of standard thickness as specified for 125 lb. per sq. in. working pressure. Butt welded steel pipe was used on intermediate pressure lines 1½ in. and smaller, and on low pressure lines 3½ in. and smaller.

Table 1 gives a summary of the materials used in the three classes of pipe lines.

The Pipe Joints

Flanges for all classes of pipe lines are forged steel, all joints being flanged for pipe lines 2 in. and larger. Van stone flanges are provided on high pressure lines 2½ in. and larger, and on intermediate and low pressure lines 4 in. and larger. High pressure lines 2-in. and 2½-in. pipe size and intermediate pressure lines 2 in. to 3½ in. inclusive are screwed, flanged and welded. Low pressure lines 2-in. to 3½-in. pipe size are provided with screwed steel flanges, but not welded. All lines 1½ in.

TABLE 1—PIPE LINE MATERIALS

CLASS	STANDARD	TEMPERATURE	PIPE	FITTINGS	VALVES	BOLTS	GASKETS
High Pressure.....	600#	725 F.....	Seamless Steel Tubing	Cast or Forged Steel	Cast or Forged Steel with Special Alloy Steel Trim	Special Alloy Steel	Copper Plated Steel
Intermediate Pressure	250#	400 to 650 F	Open Hearth Steel	Cast or Forged Steel	Cast or Forged Steel with Special Alloy Steel Trim	Special Alloy Steel	Copper Plated Steel
Low Pressure.....	125#	Up to 350 F.	Open Hearth Steel	Cast Iron	Cast Iron, Bronze Trim	Standard Machine Steel	Fiber or Red Rubber

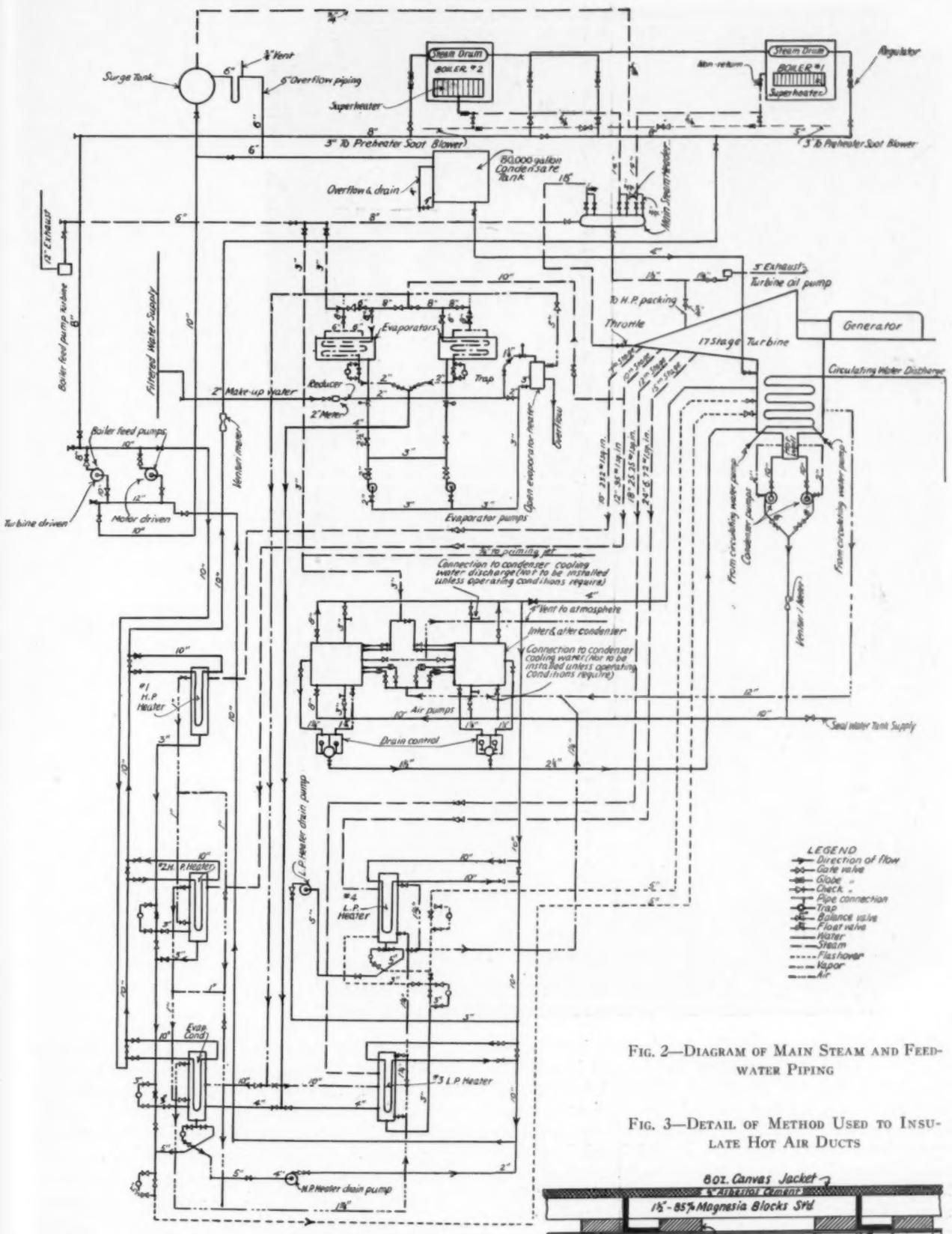
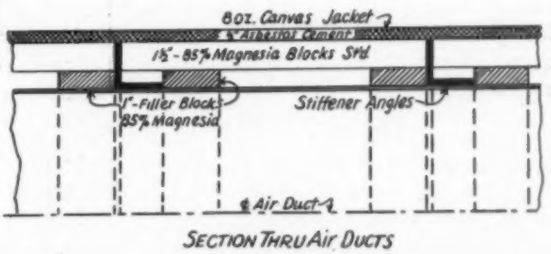


FIG. 2—DIAGRAM OF MAIN STEAM AND FEEDWATER PIPING

FIG. 3—DETAIL OF METHOD USED TO INSULATE HOT AIR DUCTS



SECTION THRU AIR DUCTS

and smaller are provided with screwed unions and fittings, the joints on the high pressure lines being screwed and welded.

Bolts for High Pressure Lines of Special Steel

Bolts for the high pressure and the intermediate pres-

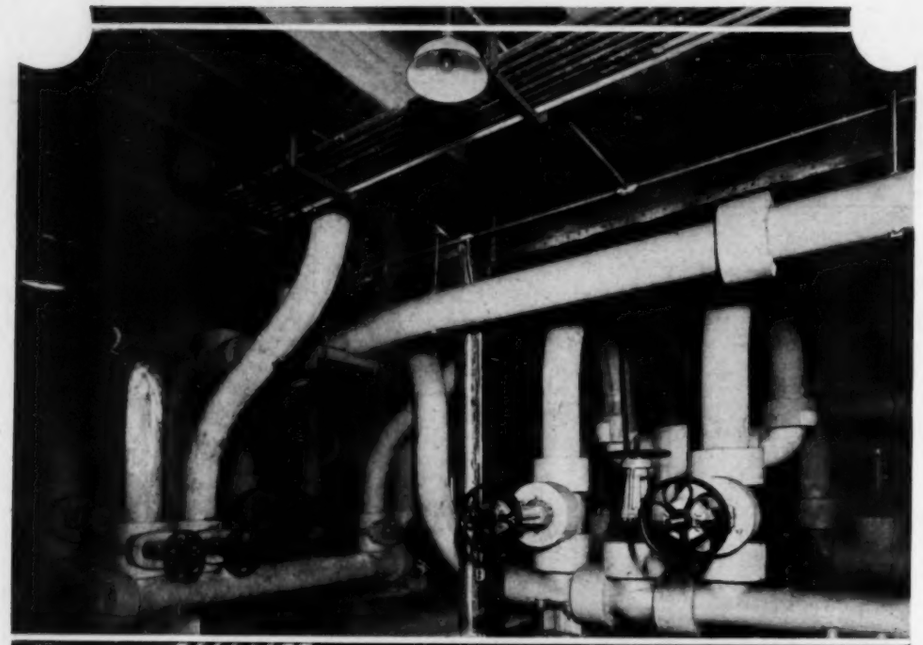


FIG. 4—FRONT VIEW OF FEEDWATER HEADER FOR HIGH PRESSURE HEATERS SHOWING HOW BY-PASSES ARE ARRANGED FOR ALL HEATERS

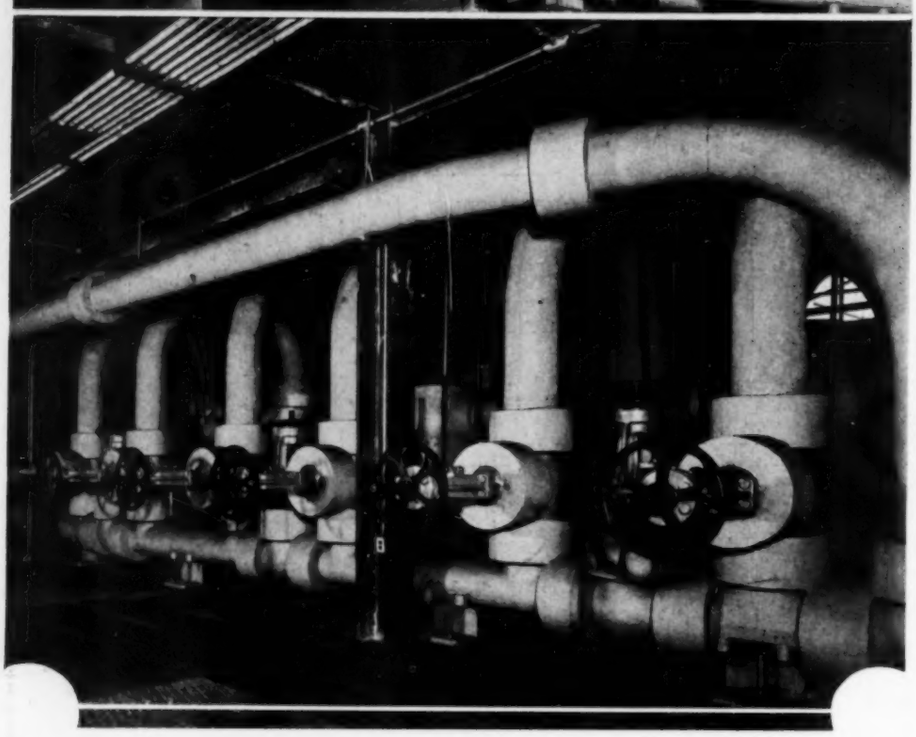


FIG. 5—VIEW OF PIPING ON ELEVATION 762.0' FLOOR SHOWING HIGH PRESSURE FEEDWATER HEADER AT RIGHT, WITH FEEDWATER PUMP SUCTION AND DISCHARGE HEADERS ON LEFT

sure pipe lines were specified to be of special heat treated alloy steel for high temperature service, the bolts to be threaded their entire length and furnished with special alloy steel nuts for each end. Bolts for low pressure lines were specified to be of standard machine steel with square heads and cold pressed, semi-finished, hexagonal nuts. Where bolts larger than 1 in. were required on low pressure lines, stud bolts were specified rather than standard machine bolts, the studs to be threaded their entire length and be furnished with two nuts.

Flange Finish and Gaskets

The selection of the type flange finish and gasket to be used on the high pressure and intermediate pressure lines, as might be expected, proved an interesting problem. It was decided to use serrated flanges, the actual specification for the flange finish being a fine phonographic finish with the depth of the grooves to be not over 1/64 in. Special copper plated steel gaskets were specified for use in such joints, and in order to insure absolutely tight joints the thickness of the copper plating was specified to be not less than .015 in.

The Valves and Fittings

All fittings and valves for high pressure and intermediate pressure lines are either forged or cast steel. Standard cast iron fittings and valves are used on the low pressure lines. High pressure and intermediate pressure valves are furnished with special alloy steel discs and seats as recommended by the manufacturer for the service required.

A special alloy steel having a tensile strength of

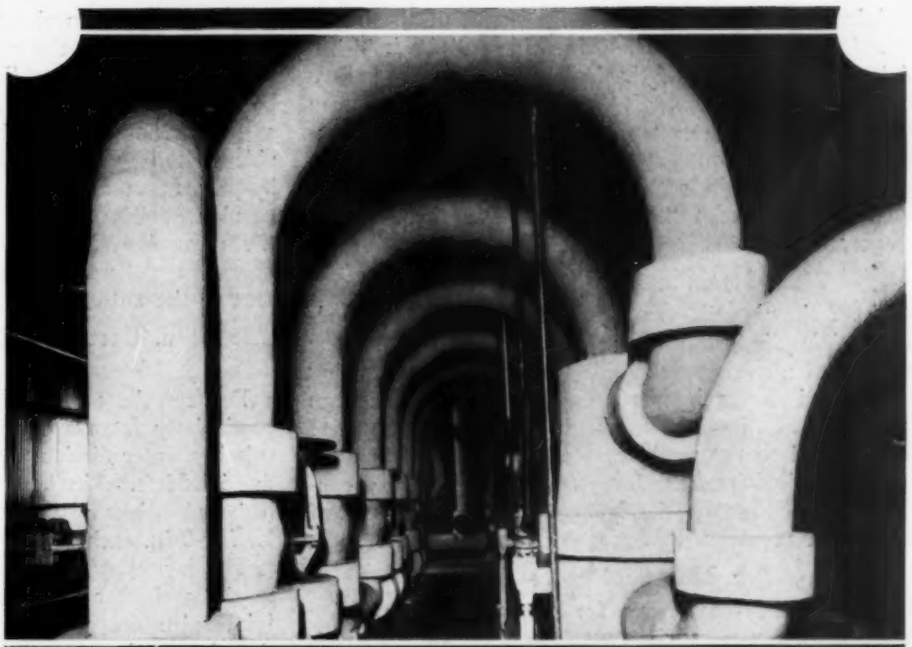


FIG. 6—END VIEW OF HIGH PRESSURE HEATERS AND FEEDWATER HEADER, SHOWING EVAPORATOR CONDENSER IN THE FOREGROUND; ALSO THE OPERATING AISLE FORMED BY THE EXPANSION LOOPS BETWEEN THE FEEDWATER HEADER AND THE HEATERS

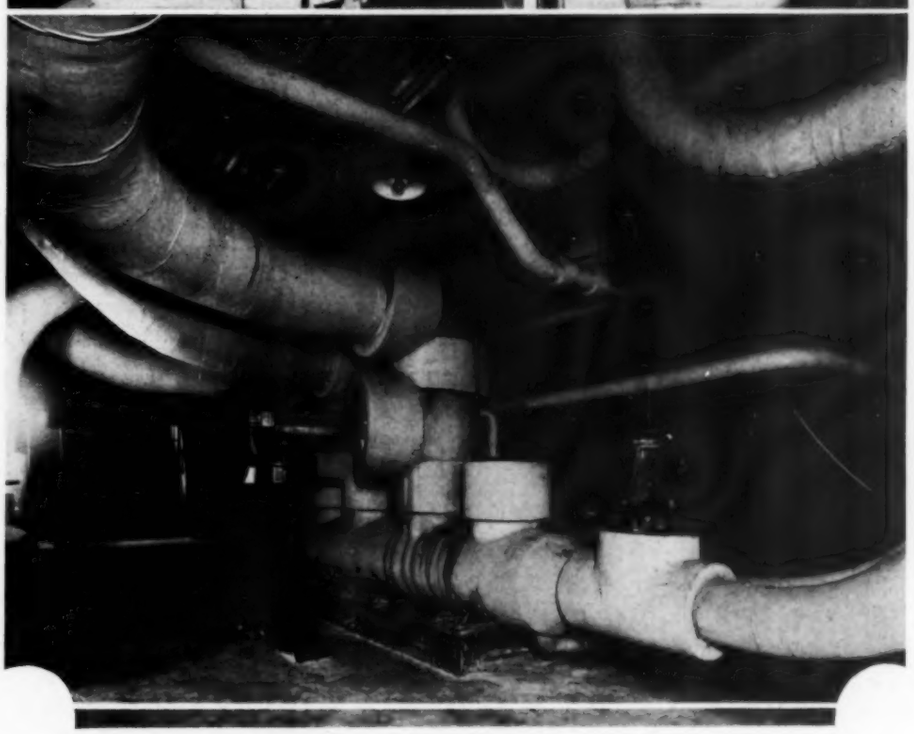


FIG. 7—VIEW OF MAIN STEAM MANIFOLD. THE AUXILIARY STEAM LEAD LEAVES THE END OF THE MANIFOLD. THE 18-IN. MAIN STEAM LEAD TO THE TURBINE IS SHOWN IN THE FOREGROUND WITH ITS MOTOR OPERATED SHUT-OFF VALVE. THE TWO 14-IN. LEADS FROM THE BOILERS ARE SHOWN ENTERING THE MANIFOLD IN THE BACKGROUND. THE BLANKED OFF CONNECTION ON THE MANIFOLD IS FOR THE FUTURE CROSS-OVER TO THE SECOND UNIT

100,000 lb. per sq. in. and used in many of the high pressure valves has the following composition: carbon, 0.25-0.35; manganese, 0.67-0.75; silicon, 0.35-0.45; chromium, 0.65-0.95; and nickel, 1.75-2.25 per cent.

Comparatively few valves were specified to be equipped with by-passes, this special feature being limited to high pressure valves 8 in. and larger, and intermediate pressure valves 10 in. and larger. There are three motor operated valves in the plant, two of these being the large 42-in. gate valves on the circulating water pump discharges, and the other being on the 18-in. main steam line to the turbine.

Responsibility of Piping Contractor

In order to simplify the purchase of materials and to concentrate the responsibility all materials except valves were furnished, delivered and erected by one contractor. The valves were purchased separately in order to allow a careful and satisfactory selection of each from competitive propositions. But with the exception of the valves, the piping contractor was held responsible for the fabrication, inspection, delivery and erection of all materials, including piping, flanges, bolts, nuts, gaskets, fittings, hangers and other supports.

Piping Gallery Simplifies Lay-out

Fig. 1 shows a typical heat balance diagram for the plant, and indicates the principal plant equipment with the sequence in which it is arranged with reference to steam and water flow.

In order to simplify the piping required, and in order to insure adequate room, the plant was designed so that

a piping gallery extends the entire length of the building. This gallery is 20 ft. -0 in. wide and 16 ft. -0 in. high, located in rear of the boilers between the boilers and the turbine room. It is below the turbine room floor and under the evaporator room. This piping gallery is as nearly centrally located to all equipment as it is possible to obtain, and provides easy access and short length pipe runs to the boilers, turbine throttle, and evaporators.

Main Steam and Feed Water Piping

Fig. 2 shows a diagram of the main steam and feedwater piping, indicating also the size of the lines and the location of all valves required. The main steam piping is arranged so that both boilers feed into a main steam manifold out of which leads are taken to the turbine throttle and to the auxiliary steam header. The main steam manifold is of the forge and hammer welded construction. Either of the boilers can be taken out of service for repairs without interrupting the service.

Provision for Future Units

Provision has been made for a cross-over connection between the present main steam manifold and the manifold for future Unit No. 2. The plant is designed for an ultimate of four units, and although it is probable that under normal conditions each unit will operate entirely independent of the others, flexibility is still a desirable factor and can be obtained through the cross-overs between main steam headers.

No By-passes Around Feedwater Regulating Valves

The principal features of the condensate and feedwater piping are, first, the provision of by-passes on all feedwater heaters so that any heater may be taken out of service for repairs without disturbing any other part of the feedwater cycle; and second, the elimination of by-passes around the automatic feedwater regulating valves at the boilers. Feedwater enters each boiler on both ends of the drum. Automatic regulators and pressure governors are installed in each line, and considerable saving was made possible by the elimination of by-passes around this equipment. Should the automatic regulator fail to function properly, it is planned to prop the valve into its wide open position, and manually operate a globe valve which is located in the line adjacent to the automatic regulator.

The 72-in. Cast Iron Circulating Water Discharge Line

Perhaps the outstanding feature of the entire plant, particularly from a piping standpoint, is the 72-in. circulating water discharge line from the condenser. This line is 185 ft. -0 in. long, and is constructed of cast iron bell and spigot pipe delivered in standard lengths of 12 ft. -0 in. each.

The laying of this pipe line was very difficult, due to the weight of each section. Making up the pipe joints also proved to be a task due to the fact that the pipe sections were not true to size or shape. The diameter in some cases varied one inch plus or minus from the exact dimension required, and in addition to this the ends were not square with the pipe.

The joints were made up with braided jute, rubber V-shaped sealing rings, and lead wool. However, due to the variation in dimensions for the different pipe sections, which resulted in large uneven gaps between the spigots and the shoulders on the bells at some of the joints, a satisfactory joint could not be obtained, since the jute when packed came all the way through into the inside of the pipe. To overcome this condition and to

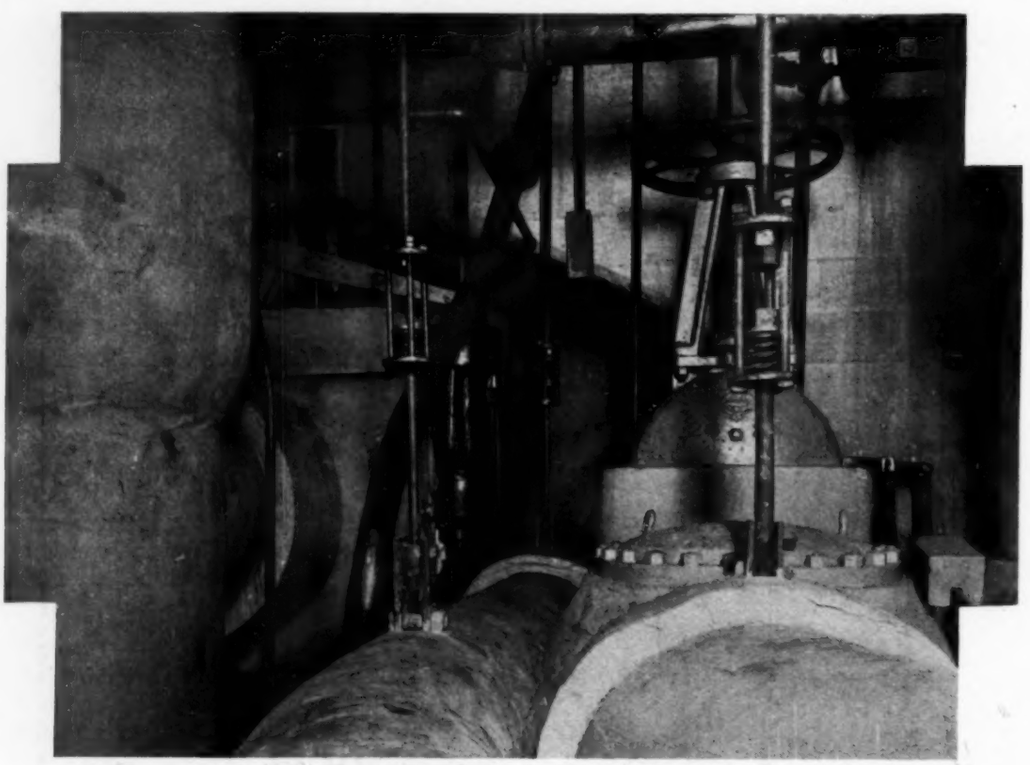


FIG. 8—VIEW OF PIPING UNDER THE TURBINE, SHOWING THE MAIN STEAM LEAD AND THROTTLE VALVE

insure air tight joints, a portland cement sand mortar, using about a 1:2 mix, was trowelled into the joint from the inside of the pipe. This mortar was applied fairly dry and packed in, the purpose being to hold the joint materials in place.

This line represented a considerable saving over the conventional method of discharging the circulating water through large underground concrete tunnels extending the entire length of the plant.

Stresses in Piping Thoroughly Analyzed

When high temperatures are encountered such as is the case in the high pressure steam lines in this plant, expansion and the proper provision for it presents a major problem.

In order to provide a satisfactory installation it is essential: first, to avoid excessive moments at the joints, in order to prevent damage to them and subsequent leaks; second, to avoid excessive stresses in pipe; and third, to avoid excessive reactions which would damage anchors, guides, or equipment. In designing the high pressure piping for this plant the design and layout was not calculated from empirical rules, nor was it left to the judgment of the designer, but a thorough analysis was made of the line with regard to elongation, wall stresses and reactions. In making these analyses the calculations were based on the paper presented by W. H. Shipman at the annual meeting of the *American Society of Mechanical Engineers* in December, 1928.

The length of each steam line was estimated and the expansion calculated for the temperature range to be encountered, all calculations being based on data contained in the paper referred to above. Bends and loops were then provided in the lines to take up the required expansion, the number of bends, and the minimum radius of each, being such as to keep within the maximum wall stresses allowable in the pipe. All main steam lines were securely anchored at the points where connections to equipment were made. Movement of the pipe was forced in the desired direction through proper location of guides. All high temperature lines, such as the main steam from the boilers to the header, and from the header to the turbine throttle, were supported throughout the entire length by spring supports except at points of anchorage, giving freedom of movement in all directions.

It is a significant fact that expansion was adequately provided for, and at the same time special fittings, special short radius bends, and special mechanical expansion joints were eliminated.

Study Given to Insulation of Piping and Ducts

Considerable thought and study was given to heat insulating materials for the plant piping and equipment in an effort to secure a workmanlike job with the required insulating efficiency for the least cost possible. A study was made of each line, heat losses being calculated for various thicknesses and types of insulating materials, in an effort to determine the most economical thickness to use. However, it must be stated that the most economical thickness was not always the one selected; very often it was found necessary to increase this thickness in order to secure a comfortable room temperature.

The insulating materials specified for various lines and the thicknesses used are shown in the tabulation below, Table 2.

Special high temperature insulating material was specified on the high pressure steam lines, and the first extraction steam lines, the temperature in these lines varying from 600 F to 725 F. Also these lines were specified to be covered with two layers of blocks. This will be noted from the table, the inner layer being specified to be high temperature blocks, and the outer layer to be 85 per cent magnesia blocks.

All joints were specified to be pointed up and the outer layer to be trowelled smooth. On all air ducts and equipment such as heaters, evaporators, etc., the insulating blocks were covered with a ½ in. layer of asbestos cement trowelled to a smooth finish. This is

TABLE 2—INSULATING MATERIALS

LOCATION	TEMPERATURE F	THICKNESS OF INSULATION IN INCHES			
		HIGH TEMP. INSULATION	85% MAGNESIA	CEMENT	TOTAL
Hot Air Ducts & Breeching	375—500	None	1½	½	2
Gas Duct—Leaving Boiler	500—780	1	1½	½	3
Main Steam	725				
½" to 1½" Pipes	.....	Std.	Std.	None	D'ble Std.
2" to 3" Pipes	.....	Std.	Std.	None	D'ble Std.
4" to 10" Pipes	.....	1½	1½	None	3
12" and Larger	.....	1½	2½	None	4
First Extraction Steam	600				
½" to 1½" Pipes	.....	Std.	Std.	None	D'ble Std.
2" to 3" Pipes	.....	Std.	Std.	None	D'ble Std.
4" to 10" Pipes	.....	1½	1½	None	3
12" and Larger	.....	1½	1½	None	3
Second Extraction Steam	450				
½" to 1½" Pipes	.....	None	1½	None	1½
2" to 3" Pipes	.....	None	1½	None	1½
4" to 10" Pipes	.....	None	2	None	2
12" and Larger	.....	None	2	None	2
Third Extraction Steam	310				
½" to 1½" Pipes	.....	None	Std.	None	1½
2" to 3" Pipes	.....	None	1½	None	1½
4" to 10" Pipes	.....	None	1½	None	1½
12" and Larger	.....	None	1½	None	1½
Fourth Extraction Steam	175				
½" to 1½" Pipes	.....	None	Std.	None	1½
2" to 3" Pipes	.....	None	Std.	None	1½
4" to 10" Pipes	.....	None	Std.	None	1½
12" and Larger	.....	None	Std.	None	1½
Condensate	150—250				
½" to 1½" Pipes	.....	None	Std.	None	1½
2" to 3" Pipes	.....	None	Std.	None	1½
4" to 10" Pipes	.....	None	Std.	None	1½
12" and Larger	.....	None	Std.	None	1½
Feedwater	250—400				
½" to 1½" Pipes	.....	None	Std.	None	1½
2" to 3" Pipes	.....	None	1½	None	1½
4" to 10" Pipes	.....	None	2	None	2
12" and Larger	.....	None	2	None	2
Heater #1	600	None	2	½	2½
Heater #2 and Evaporator	450	None	2	½	2½
Evaporator Condenser	275	None	1½	½	2
Heater #3	310	None	1½	½	2
Heater #4	175	None	1½	½	2
Water Wall Headers & Pipes	400	None	2	None	2
Inter & After Cond	250	None	1½	½	2
Surge Tank	250	None	1	½	1½

necessary in order to secure an even surface and a workmanlike job.

Insulation Has Canvas Jacket

Previous experience with similar jobs led to the decision to require all insulation, including that on the air ducts, heaters, evaporators, and all other equipment, to be enclosed in a canvas jacket pasted and sewed in place. This represents an added expense, particularly on the air ducts and equipment, where the exposed surfaces represent large areas, but past experience has made the extra cost appear to be more than justified. If these surfaces are not encased in a jacket it is practically impossible to keep the asbestos cement from cracking and falling out of place, or from being knocked loose by workmen or other operators in the daily routine of plant operation and maintenance.

Insulation of Ducts of Particular Interest

One particular feature of the actual installation of the insulating materials is especially noteworthy. This is the method used to insulate the hot air and gas ducts. In order to secure the required stiffness it was necessary to provide stiffener angles at regular intervals along the surfaces of the ducts. These stiffener angles make the heat insulation erector's job a difficult one where an effort is to be made to produce a neat and pleasing appearance to the finished work.

Fig. 3 shows a detail of the method used in insulating all air and gas ducts. The insulation was blocked up and raised so that the outside of the insulation was flush with the outstanding legs of the stiffener angles and flanges. This resulted in a smooth surface along the entire surface area of the ducts. Raising the insulation also left an air space around the entire perimeter of the ducts, and this is expected to increase the effectiveness of the insulation.

All pipe lines, whether insulated or uninsulated, are painted light gray with the exception of oil lines, these being painted dark green. No color scheme is used to designate various steam and water lines.

Cost of Piping

The total cost of all piping, including fittings, valves, bolts, gaskets, insulation materials, supports, guides, and anchors, amounts to slightly less than four dollars ($4.00) per kw. This, however, does not include the large 72 in. dia. circulating water discharge line.

Applications for Metallic Gaskets

By E. W. Sylvester*

IN an article on joints for oil lines published in the December Heating, Piping and Air Conditioning reference was made to the use of metallic gaskets in making up flange unions of oil lines. The following notes refer to certain practical points in connection with the use of metallic gaskets in piping systems which have come under the observation of the writer.

Blowing out of gaskets in flange connections and valve bonnets in a 2-in. hydraulic (oil) power piping system was a source of continual grief to the installing engineers. The working pressure in this system was 750 lb. per sq. in., and the system was subjected to continual vibration and wide and sudden pressure fluctuations. This problem was finally solved by the use of copper-asbestos gaskets. These metallic gaskets were installed in flanges of the recessed type having male and female faces and have proved entirely satisfactory in this severe service.

Considerable difficulty has been experienced with certain shipboard fuel oil piping systems having flange joints made up with non-metallic gasket material due to rapid deterioration of the gaskets. Some operating engineers who have been shipmates with these systems solved this problem by substituting hand made sheet copper gaskets whenever it became necessary to renew defective gaskets. It would seem that commercial metallic gaskets are specifically indicated for use in making up flange unions in fuel oil piping systems and that an extension of their use in this service both afloat and ashore is justified. In numerous shipboard fuel oil piping systems flanges with ground faces have been used to eliminate the use of gaskets and subsequent leakage troubles due to gasket failures. A variation of the metal to metal flange joint which has been successfully used is a joint one face of which carries several narrow circumferential ridges which bear on the flat machined face of the partner flange. These machined ridges are in effect "built-in" metallic gaskets. The use of commercial metallic gaskets in plain machined flanges would appear to offer a satisfactory joint at a material saving in cost as compared to the metal to metal joints described above.

A type of metallic gasket which appears to be particularly useful in making up joints with flanges that are not quite true is the corrugated metal gasket made up of a single sheet of metal with the corrugations on both sides filled with asbestos cord. Due to the flexibility of this type of gasket the writer has succeeded in making up tight joints with it in cases where the flanges were distorted and tight joints could not be obtained with less flexible metallic gaskets.

On plain unrecessed flanges the writer has used in test installations metallic gaskets covering the faces of the flanges and punched out for the flange bolts and also relatively narrow ring type metallic gaskets the outside diameters of which were such as to provide a small clearance between the outside edge of a gasket and the flange bolts prior to bolting up. These gaskets were used on a 4-in. line of which the test pressure was 132 lb. per sq. in. The narrow gaskets lying entirely within the bolt circle were satisfactory as the joints were easily made tight as due to the relatively small area of the gaskets, little effort was required to compress them sufficiently to obtain a tight joint at the test pressure of the system.

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By Malcolm Tomlinson*

What are the differences between drying, vacuum drying, low humidity drying, evaporation, distillation, vaporization, dehumidification, dehydration, absorption, adsorption, centrifugal drying, tempering, and baking? There are many forms and methods of drying, all having their various applications. Malcolm Tomlinson's "introduction to drying" published here classifies and explains them.

An Introduction to Drying

DRYING is one of the most ancient arts. In primitive times man learned its use in making pottery and in the preservation of skins, meats and cheese. Later the Sumerians, who founded the first empire in the Tigro-Euphrates valley, developed the art in the production of excellent pottery, for the making of bricks and for the preservation of their cuneiform records on tablets and cylinders of clay. Through the past 6000 years the practice of the art has gradually improved until there are now drying methods to meet almost every drying problem. The difficulty is, of course, the determination of the most suitable method for the case in hand. This difficulty is further complicated by serious handicaps. One is that many of the crude methods of earlier ages are still practiced by the uninformed. In this case the poor quality of the dried product brings discredit to the art. As a result there is often little appreciation of the economy to be had through the refinements of modern drying.

Drying Known by Various Names

The processes for the removal of liquids from other liquids or from gases and solids are known by various names. When the liquid removed is valueless the removal of a small quantity is termed **drying** while the removal of a large quantity is known as **evaporation.** When the liquid removed is valuable the process is called **distillation.** There is no finely drawn division between these three processes and the apparatus employed in each case is often identical. The resultant confusion, in the technical literature, is greatly increased in the drying field by many more confusing terms. This is due, in a large measure, to the highly specialized developments in the mechanical and chemical divisions of drying. Some of these confusing terms will be discussed.

Evaporation as applied to caustic liquids. Distillation as applied to gasoline and benzene.

Seven Fundamental Methods

There are seven fundamental methods by which drying can be accomplished. The most important of these is the process known as **vaporization.** This consists of changing the liquid to be removed into a vapor and then of separating the vapor from the structure—whether it be a gas, a liquid or a solid—of which the vaporized liquid had been a part.

Vaporization as applied in turning water into steam.

A second method consists of turning the water content of liquids, gases and solids into ice and the removal of the ice. This method is the reverse of vaporization in that it employs condensation and cooling for its purposes and often requires some mechanical form of agitation. The drying method known by the awkward term **dehumidification** is a form of this deposition method in which cooling of air-water vapor mixtures is seldom carried as low as the freezing temperature.

A third method is the removal of water from air-water vapor mixtures or from hygroscopic solids by the **capillary action** caused by a difference in vapor pressure.

The fourth method is known as **adsorption** and consists of the removal of water from air-water vapor mixtures by surface tension. Many of the cases known to mechanical engineers as absorption, which is the third method here described, are actually cases of adsorption or consist partially of the adsorption process.

Adsorption as in the use of charcoal to remove moisture from the air.

A fifth method consists of the **mechanical separation** of water from solid bodies by means of a screw press or centrifugal force.

Mechanical separation as in separating quartz and galena, after grinding.

A sixth method consists of the abstraction of small amounts of water from gases or liquids by **breaking down the water molecule** with sodas, carbides and nitrides.

The seventh method consists of **precipitation** of larger quantities of water out of gases or liquids by the use of a chemical drier.

Precipitation as when a solution of common salt and water is subjected to a temperature of from -5 to -15 F. In such cases ice is precipitated in the crystal form.

We have seen that dehumidification, which is a process of air conditioning, is a form of deposition

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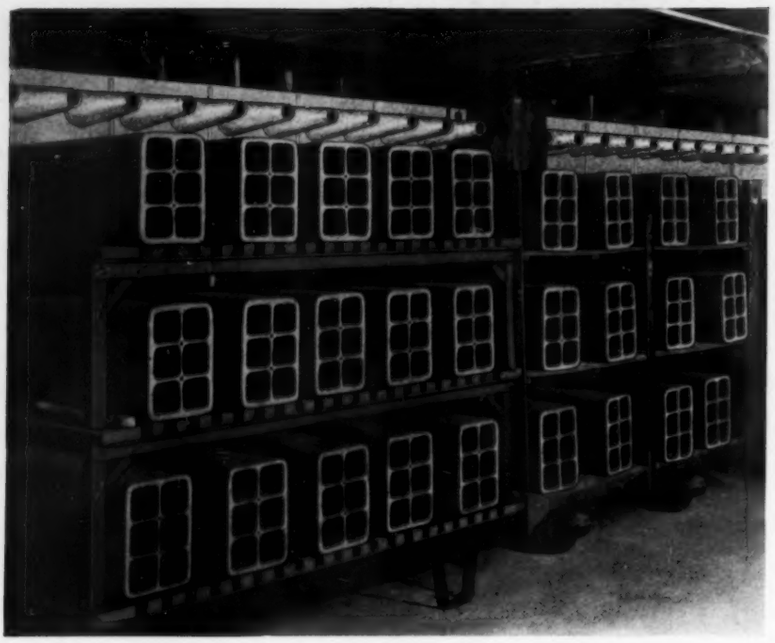

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of moisture. In recent years dehumidification has also been accomplished through the use of the adsorption method combined with the deposition method and also by the adsorption method alone.

Dehydration is a rather all-embracing term which is used for many methods of drying, evaporation and distillation. Proofing and tempering are terms used solely in air conditioning which refer mainly to delicate applications of heat for obtaining special results.

The term proofing as applied in dough baking and tempering as applied in drying grain.

Baking, on the other hand, involves handling a colloidal or an animal structure. In the first case fermentation must precede drying and air conditioning processes. In the second case a surface drying, or searing, precedes the actual baking. Therefore, in a strict sense, drying is only a process in baking. As far as air conditioning is concerned it is evident that drying can only be practiced through lowering the relative humidity, by dehumidification or by using very low relative humidities.

The various forms of mechanical drying may be classified on the basis of air pressure. Where this pressure is natural or atmospheric, drying consists of dehumidification, low humidity drying and drying at high, ordinary and low temperatures. Vacuum drying is secured by the maintenance of air pressures below the atmospheric, while high pressure drying involves the use of pressures above the atmospheric.

Selecting the Drying Method

The selection of a suitable form of drying must depend considerably on the temperature which the substance to be dried can stand without deterioration in structure, color or appearance. If heat has no particular effect, from a quality standpoint, on the substance dried temperatures of from 212 to 1,200 degrees Fahrenheit can be employed with considerable economy, not only in the amount of heat required but also in the rate of drying. Otherwise ordinary or low temperatures must be used where the air is held at atmospheric pressure. When the time interval permitted for the drying operation must be short and the quality of the substance dried is affected by ordinary or low temperatures the vacuum form of drying is advantageous. Here the reduced air pressure depresses the boiling point of the liquid to be removed. By this means, the rate at which the liquid is vaporized is greatly increased and somewhat higher drying temperatures may be used without any danger to the substance. High pressure drying may be accomplished by means of screw or centrifugal action at that temperature which is most suitable. Centrifugal action may be had with the centrifuge, or hydro-extractor, such as is used for squeezing water out of clothes in a laundry or with the more elaborate centrifugal compressor which is used for drying rare gases and air.

Air Movement Important in Drying

So far, the forms of drying discussed have been considered as though temperature and air pressure regulation were alone necessary to accomplish the purpose intended. Actually few forms of drying are of value which do not take air movement into consideration for, even in the vacuum drier, the rate at which air is moved is of primary importance. In fact most crude forms of driers are usually deficient in this factor. Therefore, where vaporization takes place, air furnishes the one vehicle by which the vapor is removed so that the rate at which the air moves, the direction in which it is moved over, through or around the substance dried and the relative humidity of the entering air require the utmost consideration.

Drying Factors, Temperature, Relative Humidity, Air Motion, Air Pressure, Vapor Pressure, Boiling Point and Heat Supplied

The necessity for some type of control of the relative humidity in various forms of atmospheric drying

has already been noticed. This is specially desirable where the substance dried, whether of animal, vegetable or mineral origin, is hygroscopic in structure or becomes hygroscopic in the drying process. These substances lose or regain moisture from the surrounding air-water vapor mixtures very rapidly when the relative humidity is lowered or increased. It is this principle which makes dehumidification and low humidity forms of drying so effective in delicate drying operations. Both of these forms require the use of refrigeration or adsorption, but low humidity drying takes advantage of the remarkable drying qualities secured through relative humidities of from ¼ to 10 per cent. It is thus evident that there are four factors—temperature, relative humidity, air motion and air pressure—which are of vital import in drying.

Drying depends on other factors in addition to those just named. These are the vapor pressure, the boiling point and the heat supplied. The effect of these factors needs to be clearly understood.

Vapor Pressure

Every liquid has a definite vapor pressure which varies with the temperature of the liquid and with the pressure of the atmosphere. When a liquid vaporizes it mixes with the air which, by itself, also has a definite pressure which depends on its temperature. Also, as air under natural conditions always is mixed with water vapor, even when we consider it dry, the addition of the vapor of a liquid which is not water always means that there will be two vapors in the air. Now when a vapor is mixed with air the partial pressure of each, added together, gives the total pressure of the atmosphere.

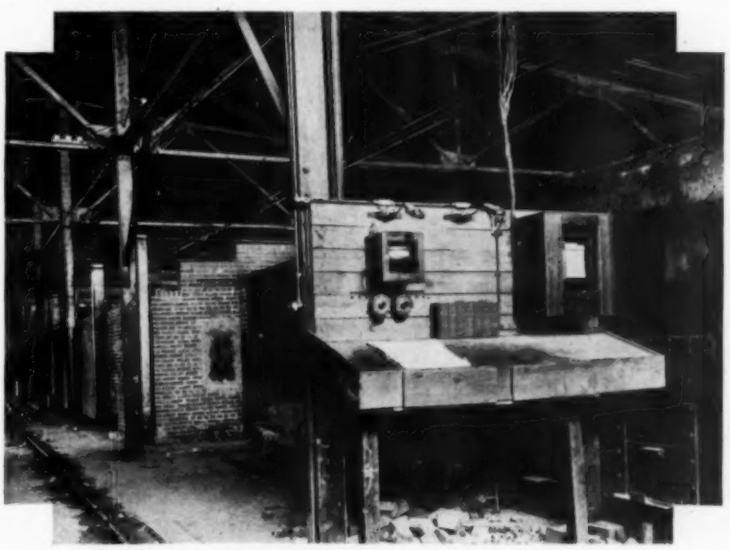
For example the pressure of saturated water vapor at normal atmospheric pressure and 100 F is 0.949 lb. per sq. in. As normal atmospheric pressure is 14.696 lb. per sq. in. the partial pressure exerted by the air alone would be the difference between the two pressures or 13.747 lb. per sq. in. When air-water vapor mixtures contain less moisture than is necessary for saturation of the mixture the partial pressure of the water vapor is reduced by the same percentage. This relative vapor pressure, which is nothing more or less than the percentage mentioned, is actually the same thing as the relative humidity.

The difference between the vapor pressure of the liquid to be removed and the partial pressure which its vapor exerts in the air governs the speed with which drying by vaporization is accomplished. This liquid is usually water and is thus in intimate contact with the water vapor in the air. As the moisture content of many substances is governed almost entirely by the relative humidity, it is clear that every change in the relative humidity is immediately followed by a corresponding change in the water content of the substance and, therefore, in the vapor pressure of the water in the substance. Thus a state of equilibrium is constantly being established. Drying requires that this effort of nature to keep up a state of equilibrium shall be constantly foiled. When the various component liquids which make up a substance are soluble in one another the total vapor pressure of the substance is reduced. Therefore any substance which is soluble in the liquid, which must be removed in the drying operation, will be harder to dry than a substance which is insoluble in the liquid. Combinations, therefore, which cause vapor pressure lowering increase the difficulties of the drying operations.

Vacuum and Boiling Point

We have already seen that a vacuum lowers the boiling point of a liquid and thus hastens the drying and also makes possible drying at higher temperatures than could be used under atmospheric pressure.

The boiling point of any liquid is that particular temperature at which that liquid, under a given pressure, will vaporize indefinitely. When we have a vapor pressure lowering, due to soluble liquids in a given substance, there is a corresponding boiling point raising. The disadvantage of the lowering therefore applies equally to the boiling point raising. The boiling point has a particular value. The greater the difference between the temperature of the drying operation and the boiling temperature for the same pressure the faster is the rate of va-



Pyrometer Equipment in a Pennsylvania Brick and Tile Plant Used for Indicating and Recording Temperatures in Tunnel Type Kilns

porization. Here is the special advantage of high pressure and of vacuum drying.

Vaporization—Absorption

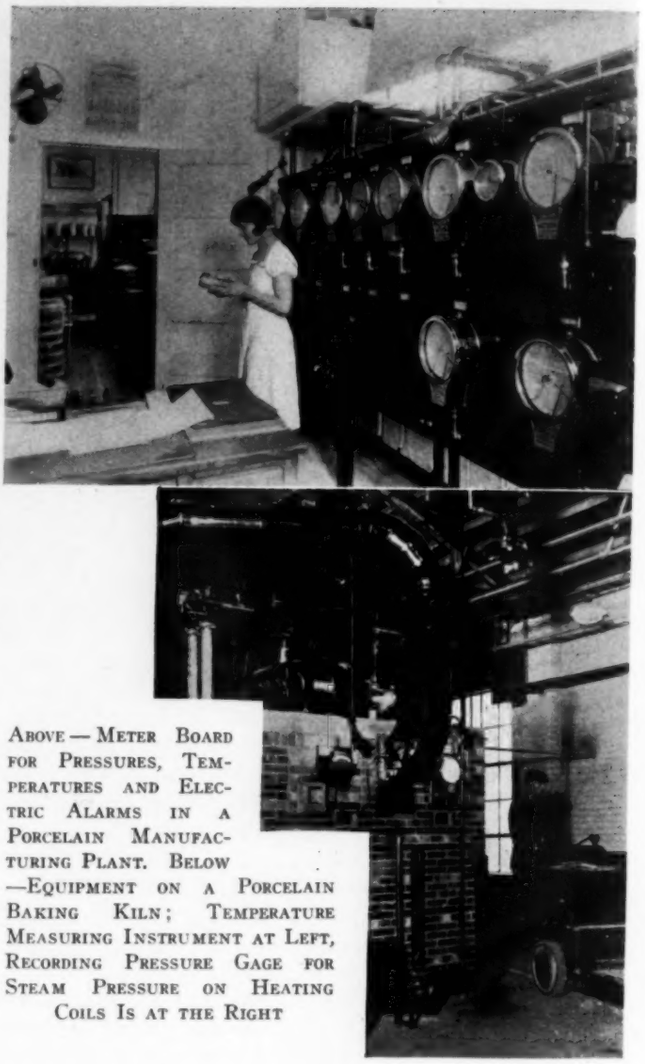
When the surface of a substance is exposed to the air, vaporization proceeds slowly through the diffusion of the liquid to the surface of the substance and through convection from the surface until a state of equilibrium is established. Continuation of the vaporization process then depends on a breakdown of the equilibrium. If the substance be contained in a closed vessel, vapor can be removed by lowering the temperature and abstracting the vapor by suction as in vacuum drying. It can also be removed by absorption in a drying medium such as sulphuric acid. As the vapor leaves the substance it carries with it heat so that vaporization causes a drop in temperature as far as the substance is concerned. Therefore additional heat must be supplied from an outside source. The amount of heat of vaporization needed varies with the substance dried and with the ratio between the pressure and temperature.

Vaporization consists of diffusion, from the interior of the substance, and surface evaporation. These two processes go on simultaneously until equilibrium is established. In some cases it is the rate of diffusion and in others the surface evaporation which controls the rate of drying. With some substances, if the surface evaporation is too rapid, the surface may become so dry that diffusion to the surface is prevented. In such cases substances dried may be permanently damaged by cracks or warpage due to the pressure exerted by the diffusing liquid. On the other hand are substances in which "case hardening" of the surface never occurs. Here diffusion is only limited by the rate of surface evaporation. This rate is governed by the velocity of the air over the surface, the area of the surface exposed and the difference between the pressure of the liquid at the surface of the substance and the partial pressure of the vapor in the air.

Heat Required

The heat required for drying operations should be figured on the basis of the amount necessary to raise the temperature of the outside air to that needed by the drier. Of course, the actual process involves heating the inside air alone to the temperature of the drier. This air must be replaced by outside air from the shop. This latter air in turn must be replaced by air from out-of-doors.

There are three methods by which heat may be supplied to the circulated air. One is through the use of any available waste heat. This method is not considered highly efficient, since condensation from the moisture in this heat often causes corrosion and also as the rate of heat transfer is quite low. Another method is the use of a closed air circuit in which the moisture-laden air is dehumidified before recirculation. The disadvantage of this method is that it requires large quantities of air towards the end of the drying period when little moisture is obtained from the drying substance. The third method, which is highly efficient in the use of heat, is to dehumidify only a portion of the recirculated



Above — Meter Board for Pressures, Temperatures and Electric Alarms in a Porcelain Manufacturing Plant. Below —Equipment on a Porcelain Baking Kiln; Temperature Measuring Instrument at Left, Recording Pressure Gage for Steam Pressure on Heating Coils Is at the Right

air and mix this dried air with the remainder of the air returning to the drier. In this way it is easy to regulate the amount of moisture in the entering air.

Air Motion

The method by which air is circulated is one of vital consideration in drying. The rate of air movement should be high where the substance dried is thin or in a finely divided state. It should be slow where the substance is thick or lumpy. A uniform velocity is essential and may be insured by the elimination of any sharp bends in ducts. The direction of flow is usually transverse where the velocity is high but often parallel for low velocities. There is no fixed rule concerning the direction of flow in relation to the substance except that the highest efficiency in sweeping the surfaces with the air should be obtained.

Rate of Drying

The danger of case-hardening, or the too rapid evaporation of surface moisture, has already been seen. With wood and ceramics this danger must be guarded against constantly. From this it is possible to see that the effect of the substance dried on the drying process depends largely on the character-

istics of the substance. Where lumber, heel-board and large sheets of paper are dried, diffusion is more rapid than surface evaporation. The rate of loss of moisture, for constant drying, is in this case directly proportional to the moisture content of the substance and inversely proportional to the thickness of the substance. In drying twine, diffusion is slow compared with surface evaporation and there is no skin effect. Here the constant drying rate is proportional to the square of the thickness. On the other hand, in drying bar soap, where diffusion is slower than surface evaporation but where there is a skin effect, the constant drying rate is proportional to the reciprocal of the weight.

Drying with Air

Drying with air is employed where it is difficult to convey heat directly to the substance dried, where the substance must be dried at a low temperature or where the substance must be dried slowly to avoid deterioration in quality. The air used must heat the substance, must transmit sufficient heat for vaporization and must take up the vaporized water. Therefore the air always leaves the drier cooler than when it entered. Since the capacity of air to take up water increases with a rise in temperature and since efficient operation requires that air leave the drier with the largest possible quantity of water, it is essential that the temperature of the air leaving the drier be as high as possible.

The heat content of the entering air must be sufficiently higher than that at the exit of the drier so that there will be enough heat to warm the wet substance and to vaporize its moisture. The less heat necessary for vaporization the less air will be needed. It therefore follows that the least quantity of air is required when it is at the highest permissible temperature as far as the substance dried is concerned. To dry by means of air successfully it is necessary to supply the right quantity of heat and of heating surface, the requisite volume or weight of air and the air pressure which will give the best results irrespective of whether it is above, at or below atmospheric pressure.

It will also be found advisable to carry out all air calculations on the basis of weight rather than volume, since changes in temperature and pressure have no effect on weight but a radical effect on volume. This precaution will effect a saving in time required for calculations.

In the selection of fans for circulation of air it is best to keep the pressure in the drier somewhat below, rather than above, atmospheric pressure because space contains more saturated vapor per unit weight of air under this condition than under ordinary or increased air pressures. The total consumption of heat in this case will also be less than in a drier operating at ordinary pressures.

Drying with Superheated Steam

Drying with superheated steam may also be used. Means should be provided, in this case, to eliminate air from the drier as quickly as possible after starting. A valve at the top of the drier will permit the escape of air and vapor at a predetermined pressure. A fan serves to provide circulation. The steam becomes saturated with water from the substance dried. Therefore a portion of the steam must constantly be removed so that the inward flow of superheated steam is not hampered. This method of drying may be used at atmospheric pressure and at pressures above the atmosphere, but it is only economical when the air is thoroughly removed, since air, in steam, lowers the rate of heat transfer.

Classification of Driers

The classification of the various forms of driers is complicated because of the great variety of combinations in use. From the standpoint of means by which heat is supplied driers may be considered under the two headings of air current and contact. The air current driers may use air from out of doors or air which is heated by radiators and gases, such as products of combustion or superheated steam. Contact driers supply heat to the substance dried by direct contact and use products of combustion, steam, hot water, oil, electricity or the latent heat of condensation. In both general classes of driers additional heat may be supplied from hot surfaces which are directly exposed to the substance. This heat is therefore radiant in character and may be solar heat from the sun or the heat from steam coils and radiators.

There is another means of classification which is based on the method of operation. Driers that are operated intermittently are always air current driers. They include loft, compartment, chamber and cabinet driers. Continuous driers of the air current type include tunnel and spray driers, while continuous driers of the mechanical contact type include drum, rotary, cylinder, and pan or agitator driers. Drum, rotary and spray driers may be built on the vacuum principle, while all of the intermittent and continuous driers are also built for operation at atmospheric pressure. In addition tunnel and rotary driers of the atmospheric types may be designed to handle air with a parallel current or counter current flow. Spray driers of the atmospheric type are also used with hot air or superheated steam.

Loft driers employ a simple form of design. They consist of rooms heated by steam coils which may be arranged in various ways. They are usually uneconomical, since the air distribution is not uniform, impairing the quality of the product. They take up considerable floor space and require frequent inspections.

Cabinet, chamber and compartment driers give a slow drying rate, which is an advantage for some substances. As the efficiency of these driers is considerably improved by continuity of operation, they are often arranged around a central exhaust or in battery formation.

Vacuum driers are used when the liquid vaporized must be saved, when the substance can not be maintained at a warm temperature in contact with air or when drying at low temperatures is required. They must be designed to withstand a pressure of 15 lb. per sq. in. from the outside or, as in rotary driers, a pressure that is equivalent to the minimum vacuum employed. The air and vapors are exhausted by

suction and the vapor is condensed in a separate compartment. The efficiency of this type of drier depends on the vacuum carried. Therefore, since the highest vacuums are obtained through the combined use of reciprocating compressors and steam jet pumps, means must be provided for supplying steam or electricity for the compressors, low pressure steam for the steam pumps and cold water for the condenser.

Tunnel driers use the principle of continuous operation in which the substance dried moves at a fixed rate through the drier. The heated air may pass the substance in the reverse direction or parallel. The counter flow principle is efficient except where case hardening is likely to occur or where a definite amount of moisture must be retained. Tunnel driers are economic of space and inexpensive as to construction.

Drum driers are used for both solids and liquids. In the case of sheets of solid materials the two surfaces alternately pass over heated drums so that very rapid vaporization is secured. The difficulty with this form of drying is when materials tend to shrink. Elongation then occurs, since the tension of the sheet on the drums is uniform, and the tensile strength of the finished product is not as uniform as when dried in a loft. This disadvantage is sometimes overcome by use of drums of varying diameters. Substances in a finely divided state of a mushy or creamy character may be dried on a single drum. In such cases the thickness of the substance is regulated on the feed side of the drier and the dried material is scraped off the drum at the end of a single revolution.

Rotary driers are used for crystalline, granular or lumpy substances where these materials do not roll into larger units. The cylinder is usually set on a slight angle so that the feed end is higher than the discharge. Rotation of the cylinder and the shelves or plates within the cylinder continually throw the materials through the hot air or gases while the charge moves slowly towards the discharge end. Some driers of this type employ a reversed current of air by passing the gases or air parallel to the charge but over the outside of the cylinder and then bringing the current back through the inside of the drier. The heat losses in rotary driers are large, even in the reversed current type, and must be an important consideration when this drier is selected.

Spray driers evaporate large amounts of water and dry the substance quickly. The liquid may be sprayed directly into a rotary drier together with the necessary heat from an oil burner or other source. In cases of a finely divided product a separator for air removal may also be necessary.

Cylinder driers are similar to the drum type of drier in operation.

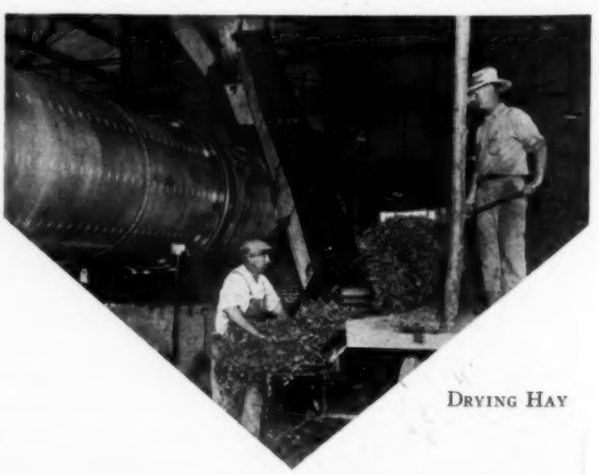
Pan or agitator driers contain agitators which constantly shift the substance dried in the bottom of the shell. The shell, the agitator or both the shell and agitator may be in motion. Hot air or gases are passed over the material and additional heat is also supplied from a jacket or through the stirring arms of the agitator.

The Drying Cycle

The drying cycle usually consists of two or more periods. One is a stage of constant vaporization and the other a stage in which vaporization is decreasing. The later period generally starts when diffusion from the interior of the substance no longer keeps the surface wet. In the constant rate period the surface is wet or saturated, due to diffusion, and the rate of drying depends on the rate of diffusion. When the heat of vaporization is supplied only through the surface of the substance the temperature of the substance is that of the wet bulb of the surrounding air. Any heat secured by radiation or conduction within the drier tends to raise the temperature of the substance above the wet bulb. This raises the vapor pressure and increases diffusion. The velocity of the air is, in this period, of the utmost importance, as it will govern the rate of drying. The next most important consideration is the supply of radiant heat within the drier. The next step in the drying period is the first portion of the falling rate stage and may be called the unsaturated surface period. The moisture is now insufficient to keep the surface entirely saturated and the drying rate can only be kept from dropping off alarmingly by increasing the air velocity and decreasing the relative humidity of the air. The next period is not influenced by the velocity of the air or its humidity, vaporization takes place mainly below the surface of the substance and the surface itself is dry. Under this condition the rate of drying falls off rapidly, as the moisture must diffuse not only through the interior of the substance but also through the surface and the surrounding gas film. Many solids omit the constant rate period. An instance of this is in lumber drying. In other materials the saturated surface period is omitted. Again, in the case of thin sheets of paper, the first two periods alone are observable.

The Drying Curve

When the moisture content of the material or substance throughout the drying cycle is plotted against the time required for drying the resultant drying curve takes the form of a parabolic curve during the first portion of the drying period and of a logarithmic curve during the last portion of the period in cases of materials in which no capillary movement of "free" water takes place. Where capillary action can take place this curve will not hold, since the drying time becomes proportional to the first power of the thickness, rather than the square of the thickness, in the early stages of drying.



Drying Hay

FIG. 1—LARGE FACTORY 165 x 235 FT., HEATED BY FOUR DIRECT FIRED HEATERS

Large Space Warm Air Heating

By O. W. Kothe*

THE architects' drawing, Fig. 1, shows a building 185 by 235 ft. in size constructed of cinder cement blocks, with many windows on each side and two saw-tooth skylights of large size, and a concrete slab roof. There is a heat loss of nearly 2,400,000 Btu per hr. at zero F.

A study of Fig. 2 shows that the office takes up part of the front. Its ceiling is about two feet lower than the factory ceiling, which enables running overhead ducts and the use of drop diffusers. There is no basement under the factory, except for the heating plant, which is a small square excavated to accommodate the four oil burning units. The inset at the upper right hand corner of Fig. 2 shows the arrangement followed for the heaters, smoke pipe and fans.

The offices are heated to 70 F and the factory to 60 F. Due to the office partitions it is necessary to return the cold air back to the fan unit as the dotted line indicates. The return air from the factory is circulated by gravity to the large iron grille in front of the heating plant where the air is picked up by the fans and discharged through the heating system.

The heater supplying the offices is thermostatically controlled so that an even temperature of 70 F is maintained. The other three units are not under thermostatic control but are equipped with controlling devices in the bonnets to regulate the oil burners and fans. Thus the fans do not start until a predetermined temperature is built up in order not to blow cold air through the system. When the oil burners are cut off and the temperature level drops, the controllers cut out the fan.

Fans of the double-end barrel type were used. Each has a capacity of about 6500 c.f.m. All told the building requires about 1,565,000 cu. ft. of air per hr. The lower inset in Fig. 2 indicates the type of heater employed, having two individual radiator drum economizers to provide long heat travel from the oil burners. Two tests were made on these heaters—the results are given in Table 1.

The heater proper is 77 in. high, 57 in. wide and has a smoke pipe collar 10 in. in diameter. Such heaters can be encased in sheet metal or brick housings. Where battery heaters are employed, as in Fig. 2, partitions are built between the heaters to enable the use of one or more units according to weather conditions.

This factory was difficult to pipe up due to the overhead shafting, pulleys, belting, drying ovens, etc.; therefore, one heat stack was taken off from each heater unit. It carries the warm air where desired and discharges it as conditions require. Two ducts carry the heated air close to the ends of the building and discharge it toward the outside wall. The other two heat stacks are short and discharge the air into the middle of the building.

All metal ducts are suspended from the ceiling, the latter being 12 ft. high; the ducts are placed at about the 11 ft. level. Shop men took care to make the Y-connections, as well as the elbows, etc., according to stand-

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TABLE 1—TEST PERFORMANCE OF HEATER SHOWN IN FIG. 2.
(FUEL USED, STRAW DISTILLATE)

ITEMS UNDER TEST	TEST NO. 1	TEST NO. 2
Oxygen in per cent	6.4	2.8
Carbon dioxide in per cent	10.7	15.04
Draft at stack in in. water	0.022	0.025
Draft in furnace in in. water	0.010	0.001
Average cold air temperature in F	98.	104.4
Average warm air temperature in F	238.	207.
Cubic feet of air per minute	2440.	5490.
Flue gas temperature in F	281.	389.4
Temperature between furnace and economizer in F	1139.	1271.
Oil fired in gal. per hr.	2.44	4.23
Heat in oil fired in per cent	100.	100.
Heat loss in dry flue gas in per cent	4.8	6.98
Heat loss due to water vapor from hydrogen in per cent	6.2	6.5
Heat extracted from economizer in per cent	24.7	21.6
Heat extracted from furnace (by difference) in per cent	64.3	64.9
Efficiency based on flue gas loss in per cent	89.	86.52
Heat content of fuel in Btu per lb	18,500.	
Specific gravity	.834	
Weight in lb. per gal.	6.97	
Delivery in Btu per hr., including heat loss from casing	280,000.	471,750.

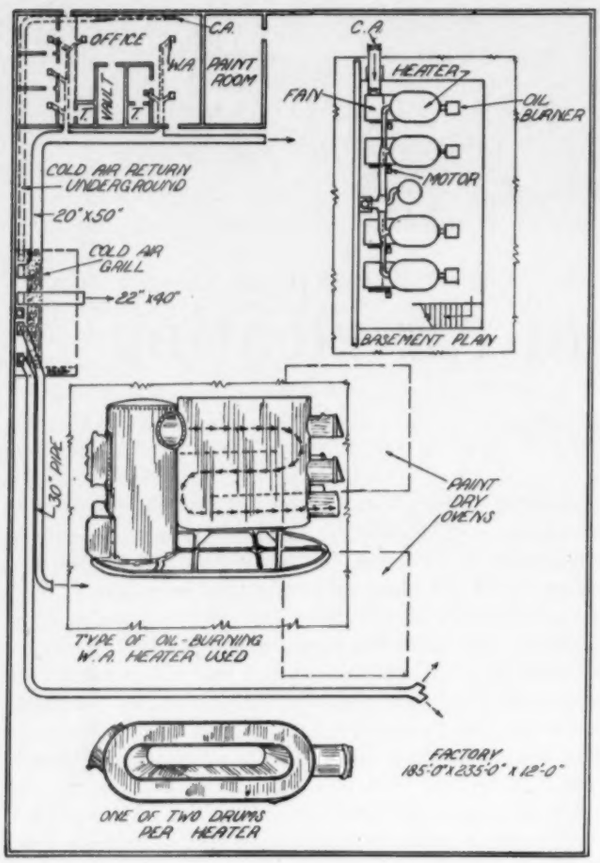


FIG. 2—TYPICAL HEATING OF LARGE FACTORY 185 x 235 FT., 12 FT. CEILINGS, USING FOUR OIL BURNING FURNACES

ard practice. The hanging was done with angle bar supports and insert rods. Twenty-four gage galvanized sheet steel was used for all duct work.

Fans for Forced Circulation

Four types of air accelerators are generally used in forced air furnace heating. The small propeller fans of two or four blades usually have capacities of 500 to about 1000 c.f.m., and may be used where the duct system is compact, of ample size and free and open to the flow of air.

Large disc fans ranging from 36 to 48 in. wheels, are finding an admirable place in many large warm air heating jobs. Often they are belt driven, to remove the motor from the air stream. Fig. 3 shows one of the possibilities in the use of such fans when employed for heating large display rooms, convention halls, etc. Any number of direct fired heaters may be built in a battery and the fans employed to accelerate circulation. One of these systems can be placed near the center of an end wall and diffuse the heated air 200 ft. or more. Circulation is automatic, the colder, heavier air finding its way back to the fan and then being recirculated through the heating system.

In Fig. 3, the heaters are placed on piers to secure better air distribution due to the large fan wheel. A good rule is to raise the heater so that the top of the fan is not higher than the middle of the fire pot; even 6 in. below center may be better.

The hood is made of galvanized steel of about 24 gage and reinforced with angle or tee bars and anchored to the ceiling. Partition walls for public buildings are often of masonry, although sheet steel is also used. Each heater is a separate unit so that one or all may be operated as weather conditions require.

As a general practice, a ¾-in. static pressure can be figured on and the fan should be selected to resist this amount of friction. Many relatively open systems will have less resistance and will require less power to operate the fan, in which case a fan with a ½-in. static pressure resistance should be provided. On extended duct systems a 1-in. to a 1½-in. static pressure is needed.

Fans must be selected for the duty they are to perform, of course, and good judgment is necessary for a designer to determine the frictional resistance set up by the system. Every type of fan has its place and no one fan can be used on every job with perfect satisfaction,

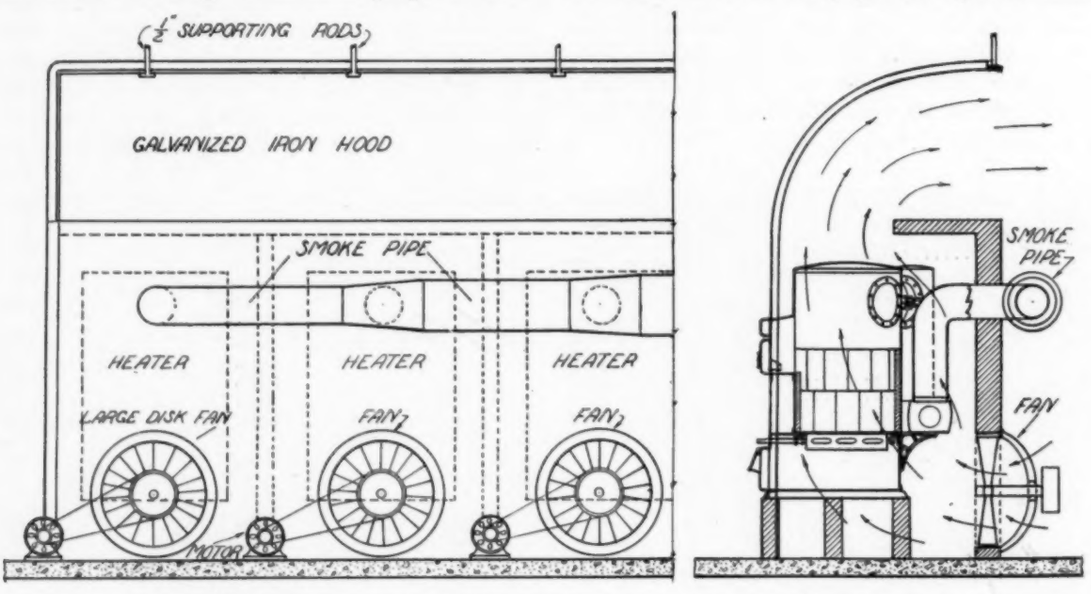


FIG. 3—AN APPLICATION OF DIRECT FIRED HEATERS TO A LARGE OPEN BUILDING

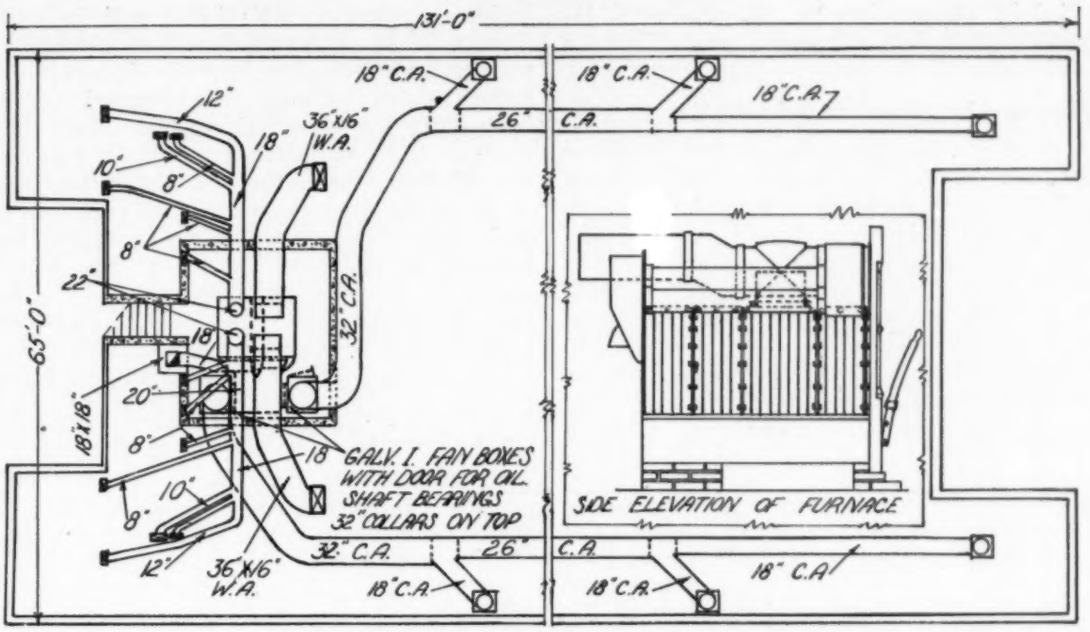


Fig. 4—Typical Heating and Ventilating Systems for Auditorium of a District School Building, Using Horizontal Direct Fired Heaters

economy and best results. Today, many special fans and appliances have been put on the market, each of which has its applications.

Figs. 4 and 5 show the application of a type of direct fired heater, two of them being used in battery form. The side elevation in the inset shows its general features. This heater weighs about 4,000 lb., possesses 7 sq. ft. of grate surface, contains 275 sq. ft. of radiating surface and has a capacity of 705,000 Btu per hr.; it will heat 6,450 cu. ft. of air per min. through 100 degrees. It can be encased with sheet steel or brick as conditions favor. The fuel may be coal, oil or gas, although if gas is used a special gas heater is preferable, since conversion jobs are not always as completely satisfactory as a regular gas heater.

In this job, the heaters are enclosed in a concrete wall. A canopy is built over the heaters with casings surrounding the heaters; warm air ducts are taken out of the bonnet. They may be round, or rectangular as shown. Numerous small stacks are distributed to lead to separate rooms surrounding the stage floor and balcony floor. These are shown in Fig. 5. All stacks, leaders and registers are standardized products selected for the heat they are to deliver. The cold air is collected in a return duct system which connects to the fan. In this case a box is built on the ends, into which the c.a.

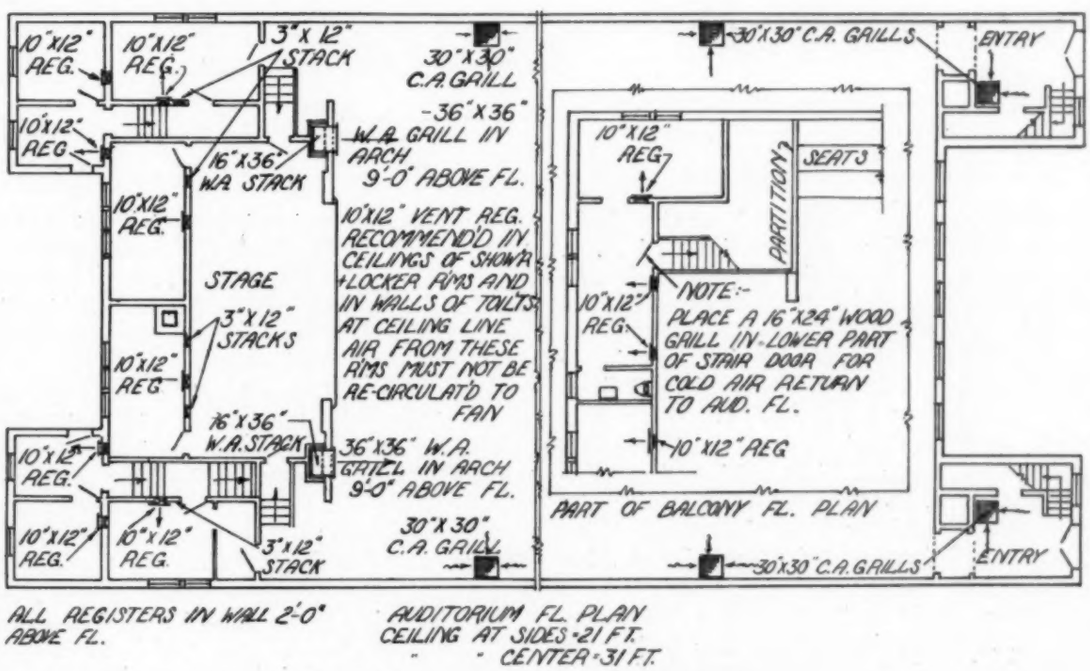

10"X12" REG.
10"X12" REG.
3"X12" STACK
30"X30" C.A. GRILL
36"X36" W.A. GRILL IN ARCH 9'-0" ABOVE FL.
16"X36" W.A. STACK
10"X12" VENT REG. RECOMMEND'D IN CEILINGS OF SHOWR + LOCKER RMS AND IN WALLS OF TOILTS AT CEILING LINE AIR FROM THESE RMS MUST NOT BE RE-CIRCULAT'D TO FAN
STAGE
3"X12" STACKS
16"X36" W.A. STACK
36"X36" W.A. GRILL IN ARCH 9'-0" ABOVE FL.
30"X30" C.A. GRILL
3"X12" STACK
30"X30" C.A. GRILLS
ENTRY
PARTITION
SEATS
NOTE:- PLACE A 16"X24" WOOD GRILL IN LOWER PART OF STAIR DOOR FOR COLD AIR RETURN TO AUD. FL.
10"X12" REG.
PART OF BALCONY FL. PLAN
ALL REGISTERS IN WALL 2'-0" ABOVE FL.
AUDITORIUM FL. PLAN
CEILING AT SIDES = 21 FT.
" CENTER = 31 FT.

Fig. 5—First and Second Floor Arrangement

ducts are led for entrance into the fan. A door is included in each box to give access to the fan bearings for oiling, repairs, etc.

Systems for Churches, Schools, Etc.

Duct systems for churches, schools, clubs, assembly rooms, etc., can be simplified in many cases. Fig. 6 shows a scheme that is applicable to many jobs. In the installation shown, oil is the fuel used. The heaters may be built in a masonry or metal wall as conditions require. The air is handled by a large disc fan which passes the returned air through an air washer to clean it and to humidify it. Then the air is passed into the furnace chamber where it is raised to, possibly, 140 or 160 F and is then discharged into the duct system.

Consider the assembly hall shown in Fig. 6 where business meetings are held; the warm air is first supplied there while the downstairs is vacant. A strong enough velocity is maintained to force the air back to the other end as much as possible. The diffuser is high to eliminate drafts. The cold air near the floor either falls down to the basement room through the grilles or down the stairways where it finds its way to the furnace fan. As the upstairs becomes warmed the return air will give up its warmth to heat the basement. Shortly before the business meetings are concluded, the dampers are switched, giving the full charge of heat to the basement room where refreshments or other forms of entertainment are often provided.

Where separated rooms require heat, as in Fig. 5, it is a simple matter to tap branch lines into the canopy and run separate leaders to such rooms.

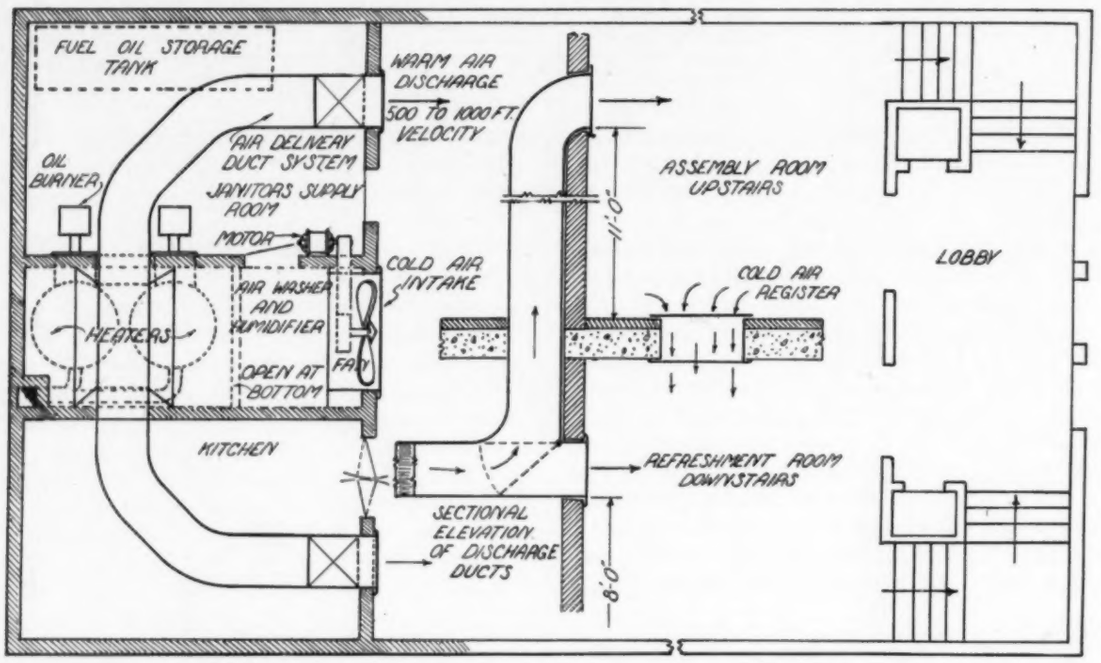


FIG. 6—TYPICAL ASSEMBLY HALL LAYOUT WITH DIRECT FIRED HEATERS

A Data Book on Gas Fuel

A copy of the loose-leaf "Data Book on Gas Fuel," which was prepared under the supervision of A. J. Evers by a committee of the *Pacific Coast Gas Association* (Frank L. Moon, chairman), has recently been received. This handbook should be of considerable value to engineers, architects and contractors interested in the use of gas in apartment houses, commercial buildings, small industrial structures, and residences.

Information is given on piping, vents and flues, gas appliances, cooking, water heating, refrigeration, laundries, and incinerators. The section on heating includes data on heat loss calculations; warm air, steam, and hot water systems; radiation calculations, and boiler selection.

The publishers are the Pacific Coast Gas Association, 447 Sutter st., San Francisco, Calif. The book is intended to be distributed by the local gas companies.

Treatise on Leather Belting

A wealth of practical information is included in the "Treatise on Leather Belting," by Professor George B. Haven and Professor George W. Swett, both of Massachusetts Institute of Technology. Included in this useful handbook is a chapter of about forty pages devoted to tables and nomographic charts.

A list of the chapter headings will serve to give a general idea of the thoroughness with which the subject has been treated. Included are sections on leather manufacture; physical properties of belting leather; methods of manufacture of leather belting; belting practice; installation and care; leather belts on motor installations; belt drive engineering; belt research; specification and inspection; and the tables, nomographs and plots.

The book is 5¼ in. by 8 in. in size, contains 250 pages, and costs $1.50. It is published by the American Leather Belting Association, New York City.

Effective Temperature—What It Means

By Edward F. Adolph*

WHEN atmospheres of varying temperatures, humidities, and air motions are compared, it is always possible to find different conditions that feel alike to the body. This is the way in which effective temperatures are derived; each is in reality a set of sensibly equivalent conditions. Further, it is found that these atmospheric conditions not only feel alike but have like physiological effects; they are physiologically effective conditions. The chief effects that have been studied in this connection[1] are changes of rate of heart-beat, changes of arterial blood pressure, changes of rectal and surface temperatures, changes of body-weight through loss of moisture, and rate of heat production by the body.

Why should all these activities of the body be influenced alike by the atmosphere? The essential connection undoubtedly lies in the fact[2] that the mean temperature of the skin surface is the same for all combinations of conditions that have the same effective temperature value. In the skin is a multitude of minute sense-spots for cold and for heat. With the body under standard conditions, then, these sense-spots furnish messages not merely to the brain so that sensations are consciously recognized, but also through the nervous system to every organ that has to do with these many bodily activities. It follows therefore that effective temperatures are the correct index to effects of the atmosphere upon all physiological functions that receive influences from the temperature sense-spots in the skin.

Effective temperatures are a true basis of reference for atmospheric conditions; but only in so far as they influence the human body through the temperature sense-spots of the skin. Obviously, further, the atmosphere cannot directly affect any part of the body except the skin and the anatomical passages through which air is breathed. It is easy, however, to demonstrate that there exist certain other effects of the atmosphere which are not correlated with effective temperatures. These effects are upon the separate processes of heat exchange from the skin, and so are not concerned with the sense-spots for temperature. Effective temperatures should, therefore, be understood to refer to activities controlled through the nervous system.

Modes of Heat Loss

So far as the evidence goes, the rate of production of heat in the body is the same at any one effective temperature, regardless of humidity or air motion.[3] This is to be expected from the fact that the processes producing heat are mainly under the control of nervous messages. The total dissipation or loss of heat from the body is likewise the same, at any given effective temperature, within the limits of error of determining it. But the several modes of heat transfer, upon which the dissipation depends, differ in rates when humidity and air motion vary, although the effective temperature is constant. For evaporation, radiation, and convection proceed at very diverse velocities in high and in low humidities, and in high and in low air motions.

If values are read from any psychrometric chart showing effective temperatures for naked men in still air along the 70 F (effective temperature) line, the data listed in Table 1 are found. Under all conditions designated as 70 F effective temperature, the mean temperature of the unclothed body surface is 95.2 F. Heat that is lost to the atmosphere by radiation, conduction, and convection must be a function of the difference in temperature between the body surface and the dry bulb. As is shown in the table, at the one effective temperature this difference spreads over a large range of values. This is particularly true for radiation, which presumably is proportional to the difference of the fourth powers of the absolute temperature of skin and of dry bulb. Heat that is lost by evaporation, on the other hand, depends likewise upon the humidity as well as the effective temperature, but in the opposite sense, for more water exudes from the body at low humidities, where the dry-bulb temperature is higher, than at high humidities. This can only mean that more water exudes from the body at low humidities, for all the water that comes to the body surface is evaporated at all humidities (below 80 F dew point), since the heat radiated from the skin is sufficient to evaporate it into an atmosphere that is already saturated. Therefore if at 70 F effective temperature

TABLE 1—ATMOSPHERIC CONDITIONS ALONG THE 70 F EFFECTIVE TEMPERATURE LINE FOR STILL AIR, FROM THE DATA OF HOUGHTEN AND YAGLOGLOU[4]. THE SKIN SURFACE TEMPERATURE IS ALWAYS 95.2 F

RELATIVE HUMIDITY %	DRY-BULB TEMPERATURE F	DIFFERENCE OF SKIN AND DRY-BULB F	RELATIVE DIFFERENCE OF FOURTH-POWERS OF ABSOLUTE TEMP. %
100	70.0	25.2	100
90	71.2	24.0	95
80	72.5	22.7	90
70	73.9	21.3	84
60	75.3	19.9	78
50	76.9	18.3	72
40	78.7	16.5	66
30	80.8	14.4	59
20	83.0	12.2	52
10	85.6	9.6	41
0	88.4	6.8	28

* University of Rochester, Rochester, N. Y.

[1] McConnell, W. J., Houghten, F. C., and Yagloglou, C. P. Jour. Amer. Soc. Heat. Vent. Engin. Vol. *30*, 199, 1924.

[2] McConnell, W. J., and Yagloglou, C. P. Jour. Amer. Soc. Heat. Vent. Engin. Vol. *30*, 457, 1924.

[3] Yagloglou, C. P. Jour. Amer. Soc. Heat. Vent. Engin. Vol. *30*, 597, 1924.

[4] Houghten, F. C., and Yagloglou, C. P. Jour. Amer. Soc. Heat. Vent. Engin. Vol. *29*, 165, 1923.

At one effective temperature the rate at which heat is dissipated from the human body is always the same. But at different humidities and velocities of air motion the ratio of loss by evaporation to loss by other modes of heat transfer varies. Effective temperature is no index to the partition of heat transfer. At certain definite atmospheric conditions, radiation, convection, and conduction are zero; at these points only the rate of cooling by evaporation is known. Evaporation is limited usually by the opposition under which water exudes to the surface of the skin in comparison to the attraction due to internal factors and to evaporation itself. It is this balance of factors that determines the slope of the effective temperature lines in relation to dry-bulb temperature lines.

the humidity is high, more of the heat being dissipated will be lost by radiation, convection, and conduction; if the humidity is low, a larger percentage of the heat will be lost by evaporation. The sum of all the losses remains the same.

Heat Loss by Radiation

It is possible to locate a series of effective temperatures and humidities where radiation, conduction, and convection are *nil*. This is done by picking the points on the psychrometric chart where the dry-bulb temperature equals the skin surface temperature. Since the skin surface temperature is always correlated with the effective temperature, there is only one such point at each effective temperature. These points are along the line of zero radiation shown in Fig. 1. The humidities at which these points are found vary enormously. At every one of them the total loss of heat from the body occurs solely by evaporation.

The conditions for zero radiation vary somewhat, of course, with the duration of exposure, since the skin surface temperature gradually increases at the higher effective temperatures.

When air motion is present, the zero radiation line changes very little. It does, however, shift slightly to the left as the velocity of motion increases, for wind increases the rate of evaporation at any given dry-bulb temperature, thus cooling the skin more. Throughout this discussion it is assumed that the temperature of the walls and other objects is the same as the dry-bulb temperature of the air.

When any one effective temperature prevails, but with the humidity higher than the value at zero radiation, then radiation, convection, and conduction take place. Since the net amount of heat dissipated from the body in a unit of time does not change unless the effective temperature changes, the evaporation is now less.

Thus while the rate of net heat dissipation is always the same at a given effective temperature, the rate at which water becomes available for evaporation does not remain the same, and radiation, convection, and conduction are not the same. All this is because the separate processes of heat transfer depend upon the dry-bulb temperature as well as on the effective temperature.

Effective temperature charts furnish no basis for separating convection and conduction from radiation. Obviously their ratio changes slightly with the dry-bulb temperature, since radiation is proportional to the difference of the fourth powers of skin surface temperature and of air temperature.

Effective Temperature Lines

One of the characteristics of effective temperatures in still air is that on a psychrometric chart each forms a straight line[4]. This means that at a given

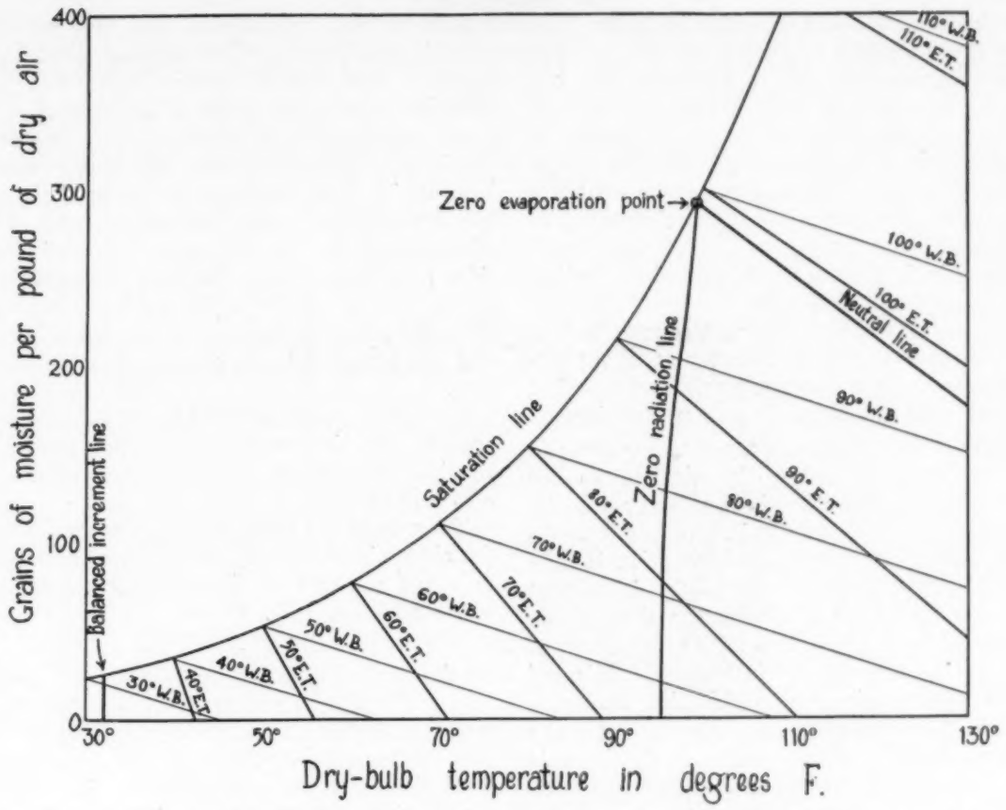


FIG. 1—PSYCHROMETRIC CHART SHOWING EFFECTIVE TEMPERATURES FOR NAKED INDIVIDUALS IN STILL AIR, TAKEN FROM THE DATA OF HOUGHTEN AND YAGLOGLOU[4].

effective temperature the net cooling from evaporation increases in proportion to the increase in dry-bulb temperature. In wind the effective temperature lines are slightly curved in some regions of the chart.

The same is characteristic of wet-bulb lines. They are straight; the excess of cooling by evaporation from the bulb over the warming by radiation, conduction, and convection increases in direct proportion to the dry-bulb temperature.

To what factors are due the differences in slope between the lines for wet-bulb temperatures and for effective temperatures? For a proper comparison, the naked human body and the wet bulb should receive the same velocity of air motion. They are therefore compared at 500 feet per minute[5], this being very near the standard velocity for wet-bulb measurements. If the human body showed the same effect of humidity that the wet bulb does, then the increment in cooling by evaporation minus the increment in all other modes of heat transfer would be alike for both objects. Actually, dryness of the air has much less cooling effect on the body than it has on the wet bulb, except at extremely high temperatures (132 F in still air, 115½ F in 500 feet per minute velocity). The slope of the lines is a measure of the advantage of dryness of the atmosphere for cooling the body when in competition with radiation, convection, and conduction. This measure may be put into figures by comparing numerically the slope of the effective temperature line and the slope of the wet-bulb temperature line, as is done in Table 2.

It is evident from the fact that greater dryness of the air gives more evaporation, that as the humidity decreases more water becomes available on the skin at the same effective temperature. The water supply to the surface is therefore not all dictated by the internal mechanisms of the body. A dry atmosphere either draws more water through the skin or

TABLE 2—RELATIVE SLOPES OF WET-BULB LINES AND EFFECTIVE TEMPERATURE LINES, AT 500 FEET PER MINUTE AIR MOTION, FROM THE DATA OF HOUGHTEN AND YAGLOGLOU[5]

Effective Temperature F	Wet-bulb Temperature at Saturation F	E. T. Slope at Saturation, Degrees D. B. per Grain of Water in One Pound of Dry Air	W. B. Slope, Degrees D. B. per Grain of Water in One Pound of Dry Air	E. T. Slope, per Cent of W. B. Slope
115	107.5	.556	.565	98
110	105.2	.436	.568	77
105	102.7	.367	.572	64
100	99.7	.353	.581	61
95	96.6	.343	.582	59
90	93.5	.285	.583	49
85	90.4	.232	.584	40
80	87.1	.180	.585	31
75	84.1	.143	.593	24
70	81.2	.102	.605	17
65	78.0	.081	.615	13
60	75.2	.050	.620	8
50	69.5	+.006	.622	+1
40	63.8	−.039	.623	−6
30	58.1	−.088	.625	−14

[5] Houghten, F. C., and Yagloglou, C. P. Jour. Amer. Soc. Heat. Vent. Engin. Vol. 30, 169, 1924.

stimulates a faster exudation of water by some local non-nervous mechanism. A certain facility to the exudation of water to the skin is given by the circulation of the blood and other functions that are controlled by the temperature sensations; the rest is due to the acceleration of the passage set up through increased evaporation. The slopes of the effective temperature lines are governed by the amount of this acceleration.

The human skin is thus seen to have the function of retarding evaporation from the body. If the skin were removed from the underlying tissues, the exposed tissue surface would lose water at about the same rate as a wet-bulb thermometer that was wetted with a one per cent solution of sodium chloride.

On every chart of effective temperatures there is one line at which humidity of the air has no net effect. Here the dry-bulb line is identical with the effective temperature line; as the humidity decreases any small increment of cooling by evaporation is then just enough to counterbalance the slight decrement to radiation, convection, and conduction that is due to the change of heat capacity and conductivity of the air. This line is denominated in Fig. 1 the *balanced increment* line. With air motion this line moves to higher dry-bulb temperatures.

Since at one effective temperature both the rate of heat transport to the skin surface and the temperature of that surface are always the same (after a standard time of exposure), the rate of heat dissipation or of net cooling is always the same. If there were no radiation, the slope of the effective temperature line compared to the vertical (Fig. 1) would be greater; if there were no evaporation the slope would be less. The actual slope is the resultant of the two. Hence, in general, the slope of an effective temperature line represents the resultant influences of increased cooling by evaporation and of decreased radiation, convection, and conduction due to the higher dry-bulb temperature. Since neither of these increments in rate of cooling is known, it is impossible to calculate therefrom the relative influences of evaporation and of the other modes of heat transfer.

Neutral Atmospheres

When the effective temperature is equal to the skin temperature, it is commonly assumed that no exchange of heat occurs between body and atmosphere. This is at an effective temperature of 99 F, termed in Fig. 1 the *neutral line*. No *net* exchange of heat does take place; but actually (in all except the saturated atmosphere) heat is coming into the body by radiation, convection, and conduction, and is leaving the body equally fast by evaporation. The existence of this balance follows from the fact that more evaporation is occurring with each decrease in humidity. The neutral temperature varies, of course, with the duration of exposure, for in this range of conditions both the skin temperature and the deep temperature increase progressively. 99 F is the neutral temperature after about 20 minutes exposure to the conditions of a body that has been previously adapted to a comfort temperature.

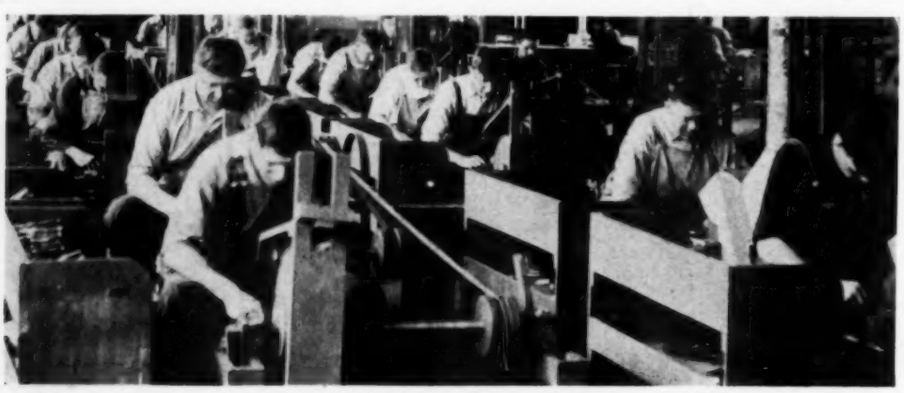
There is only one point where no net evaporation at all occurs (99 F saturated); in every other condition, whether saturated or not, either evaporation or condensation goes on.

The rate of net heat dissipation from the body (H) is the algebraic sum of the rate of cooling by evaporation (E) and the rate of cooling by radiation, convection, and conduction (C). Or

$$H = E + C$$

At the neutral effective temperature $H = 0$, and hence $E = -C$. The increment of cooling by evaporation always equals the decrement of cooling by all other modes of heat transfer, as the humidity decreases along this particular effective temperature line. At the saturated neutral temperature, not only $H = 0$, but also $E = 0$, and therefore $C = 0$. The only points at which the rate of evaporation (E) is directly known are those where $C = 0$; that is, where the dry-bulb temperature equals the skin surface temperature, which is along the zero radiation line. The only point at which the rate of radiation, convection, and conduction is directly known is where $E = 0$, which point is the junction of the neutral line and the zero radiation line. At no other points are values of C and E directly derived from effective temperature charts when combined with the measurements of rate of net heat loss from the body.

This demonstrates that it is impossible from effective temperature data alone to partition the values of H, which have been measured for each effective temperature, among the various modes of heat transfer. Nevertheless effective temperature lines help to describe the thermal properties of the human body. Certainly enough information is now available so that it can, at least, be realized that the location of every point on the effective temperature chart has a rational basis.



Heating Surface for Water Heating

Formulas and Simple Rules of Computation Reviewed

By Rollo E. Gilmore*

A READER asks (a) "How many square feet of heating surface would be required to raise the temperature of 70,000 lb. of water at 43 F to 150 F in one hour; (b) also the same amount of water from 43 F to 190 F in one hour? The heating surface is 1¼ in. copper pipe as used in a closed type storage heater; the steam is exhausted at 13 lb. absolute, temp. 205.87 F. (c) How many sq. ft. of pan (or tray) surface would be required in an open type of exhaust heater to bring about these results?"

Closed Type Heater

The formula for obtaining heating surface in a so-called closed water heater is

$$S = \frac{wr}{c(T-t)}$$

in which S is surface of copper or iron pipe in sq. ft.
W is weight in pounds of water to be heated per hr.
r is rise in temperature of the water.
t is average temperature of water in contact with coils.
T is temperature of steam.
c is coefficient of heat transmission.

The value of c for copper is assumed as 300 Btu (iron, 200 Btu) transmitted per square foot of surface per hour for each degree difference between the temperature of the steam and the average temperature of the water.

In computing the heating surface of the pipe coil it is generally customary to take the inner circumference of the steam pipe coil, as that is the real heating surface to which the heat is applied. The average temperature of the water is taken for the temperature of the water.

Solution (a)

70,000 lb. of water.
150 deg. minus 43 deg. equals 107 deg. rise in temp. of water.
107 deg. divided by 2 plus 43 deg. equals 96.5 deg. av. temp. of water.
205.87 F temp. of exhaust steam at 13 lb. absolute pressure.
205.87 deg. minus 96.5 deg. equals diff. between temp. of steam and average temp. of water.

Substituting these values in the formula we have

$$S = \frac{70{,}000 \times 107}{300 \times (205.87 - 96.5)}$$ equals 228.8 sq. ft. of copper coil.

Solution (b)

70,000 lb. of water.
190 deg. minus 43 deg. equals 147 deg. rise in temp. of water.
147 deg. divided by 2 plus 43 deg. equals 116.5 deg. av. temp. of water.
205.87 F temp. of exhaust steam at 13 lb. absolute pressure.
205.87 deg. minus 116.5 deg. equals diff. between temp. of steam and average temp. of water.

Substituting these values in the formula we have

$$S = \frac{70{,}000 \times 147}{300 \times (205.87 - 116.5)}$$ equals 383.8 sq. ft. of copper coil.

Closed tank water heaters are often selected when water is free from substances which form scale or corrode steel.

Open Type Heater

It is usually agreed that the surface area of the water carrying tray or pan is of minor importance, and no definite relation in figures for all makes (on account of the non-uniformity in shapes) can be given as practical. In the various types of open water heaters the shapes of trays or pans differ so much that any formula worked out for one would not fit another and manufacturers have met the specifications by tests of their particular designs. The tray or pan surface area used by several manufacturers for their particular designs and spacing varies from 600 to 1,500 lb. of *zero hardness* water heated per sq. ft. of tray or pan surface.

The intended function of the tray, in addition to acting as a depository for carbonates of lime, magnesia, etc. (generally termed scale), is to mix the water and steam in the most efficient manner, either by dripping or spraying the water through the steam, condensing it.

Open type water heaters have been built without trays or pans by injecting the water in through spray nozzles, a jet head in the top of the heater or through a conical opening so that water is thrown outward within the heater in the form of a thin conical sheet. However, trays or pans are more general.

With water of zero- to two-grain hardness, some definite tray surface would be based on the particular design, while if five- to ten-grain hardness, it would call for twice the surface, depending on the kinds of solids composing the hardness; from 11 to 15 grains hardness, the tray surface should be around four times that required for zero hardness.

It has been observed in the case of heaters operating at fair capacity, and with water containing scale forming solids, that when the heater has been opened up most of the scale deposit will be found on the upper trays and against the side of the heater itself in the back corner farthest from the steam inlet, the trays or pans immediately in front of the steam inlet and along the bottom of the heater being relatively free from scale. This indicates that the steam sweeping into the heater lifts the water off the trays and throws it back against the back side of the heater and that most of the heating takes place up at this point and on the upper trays.

The first condition in the problem was to heat the water to 150 deg. and the other to heat it to 190 degrees with 13-lb. absolute pressure exhaust steam at 205.87

* President, Chicago Association of Consulting Engineers.

degrees temperature. An open water heater will heat water to within four to six degrees of the temperature of the incoming steam, provided there is enough steam.

Vacuum Formed

This brings up an unusual problem of operating an open type of water heater under a vacuum. If there is not enough steam, then a vacuum will form in the heater; consequently, the only way that these temperatures could be maintained would be either to restrict the flow of steam or operate the heater under a high vacuum. It would not be an ideal installation, for it would require a large vent condenser with an air ejector of sufficient size to maintain a lower vacuum, or a pump of suitable design would be necessary, as water would be heated to approximately 200 degrees. Ordinary operation requires suitable venting.

Automatic Control

Automatic control and regulation of temperature can be accomplished by the standard types of control valves on the steam inlet with flexible tubing and the insertion thermostat bulb installed in the water storage chamber of the heater. Due to the time lag, automatic control is not entirely satisfactory for some uses.

Arrangement of Trays

The pans or trays are generally made of light metal in removable sections, and set in a slightly inclined position, first one way and then the other, so that the water dropping on the high side flows to the opposite side and to the tray below. This is to break up the water into small particles so there is a thin film or layer of water on the tray absorbing heat from the exhaust steam. The design of the tray or part should be analyzed for efficient mixing of steam and water.

The space between pans or trays is generally made not less than one-tenth the width for rectangular and one-fourth the diameter for round pans. The generally accepted practice is to use not over six pans or trays in a tier, since it is advisable to proportion the pans so as to obtain as low velocity over each pan or tray as is practicable.

Computing Capacity

In computing the capacity of the open type water heater shell it is generally accepted that the basis of calculations be upon the total volumetric content of the heater and upon the water storage content. Experience has shown that one cubic foot of heater volume, exclusive of pans or trays, filter and oil extracting devices, is ample for each 20 boiler horsepower in order to raise the temperature to within four to six degrees of the steam temperature; in other words, to obtain the cubical contents of a rectangular heater, we divide the boiler hp by 20, or the number of pounds of water heated per hour by 600, assuming 30 lb. of clear water per hp per hr.

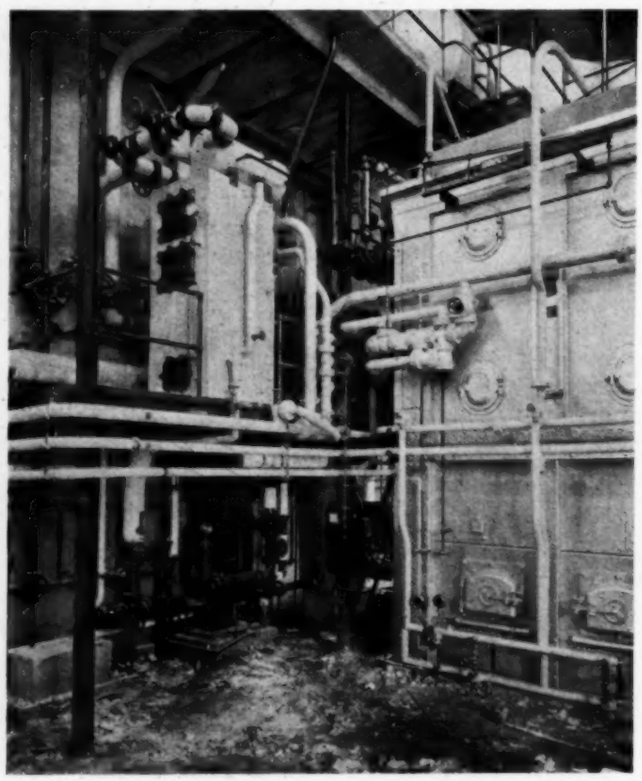
Then for proper water storage we check to determine if the heater will furnish water for any unusual demand, that we have sufficient volumetric water storage below normal water line (not overflow line) to provide a reasonable reserve capacity, generally three to four minutes, although a minimum of two to a maximum of five minutes is the practice of some engineers. It is frequently desirable, because of operating conditions, to provide the larger storage capacity.

Space for Filters

Filters, when required, occupy from ten to fifteen per cent of the volume of the shell in the horizontal type and from fifteen to twenty per cent in the vertical, the smaller percentage corresponding to designs for clear water and the larger for muddy water or water containing a considerable quantity of impurities.

It is important in the general use of the open heaters to install a vent to atmosphere for the removal of all non-condensable gases and air. Exhaust steam utilized in heating water in an open heater should always enter the heater through an efficient oil separator.

Surface Deaerating Water Heater, Showing the Piping Connections. 36,000 lb. of Water per Hour



Thus for a capacity of 70,000 lb. per hour the open water heater shell should have a minimum total volumetric capacity of approximately 116 cu. ft. After obtaining this figure, manufacturers' designs should be consulted and one of the standard types nearest meeting this requirement selected, which for this example we will say is 120 cu. ft. Next we obtain the water storage in cu. ft. to the normal water line together with approximately 25 per cent additional to the water overflow line.

Note: In practical application the figures arrived at in the above solutions are probably minimum values. Judgment based on analysis of water would determine if additional surface should be allowed to compensate for scale forming ingredients.

Some Reports on Research

Controlled Atmosphere Rooms in New Medical and Dental Research Building

University Building Provides for Air Conditioning Studies

THE Medical and Dental Research building of the college of medicine, University of Illinois, is the fourth unit of the research and educational hospitals group in Chicago. It required twenty-one separate systems of piping to take care of its complex services. Eight separate systems of ventilation have been employed, and twenty-three fans have been installed.

Granger & Bollenbacher, Chicago, are the architects for this building; Ernest V. Lippe is the consulting engineer. The architectural design was directed by Frank L. Venning, and the structural design was prepared under the direction of John O. Merrill. The building, equipped, represents a total cost of $1,500,000.

When this unit is completed, the medical group will present a quadrangular and inner court arrangement covering a ten-acre plot. The Medical and Dental Research building accommodates 700 students. It is the first unit of the medical group designed primarily for the use of undergraduate students. It houses the administrative department of the college of medicine, the departments of anatomy, physiology, pharmacology, and chemistry, and provides quarters for the college of dentistry. There will be two lecture halls seating 300 students each, one that accommodates 200 students, and seven similar rooms with 50-student capacity each.

General and Special Air Problems

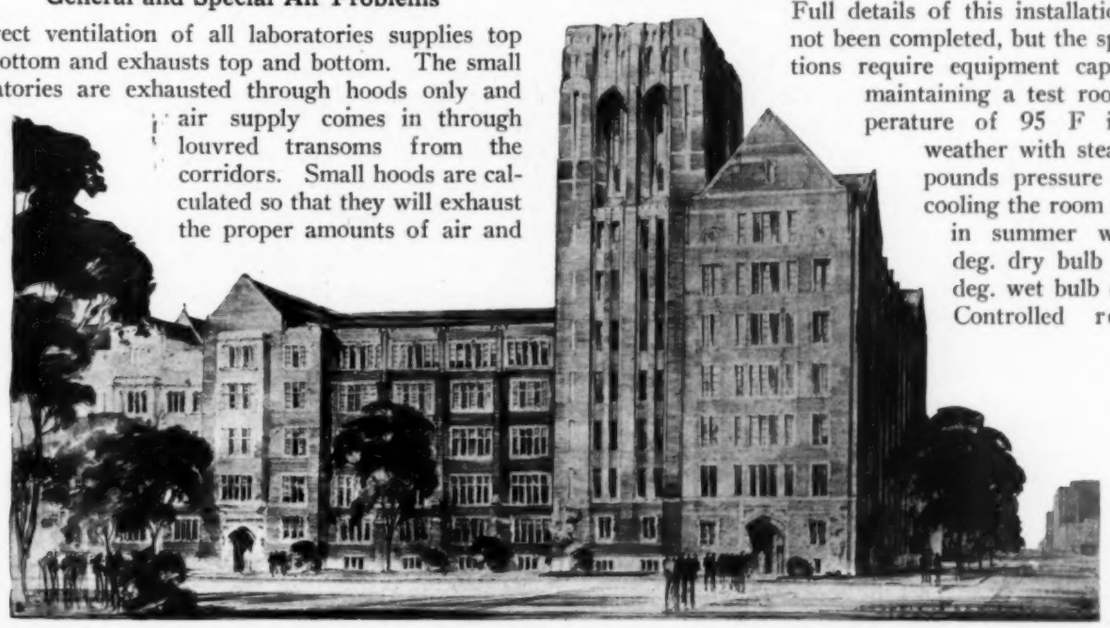
Direct ventilation of all laboratories supplies top and bottom and exhausts top and bottom. The small laboratories are exhausted through hoods only and air supply comes in through louvred transoms from the corridors. Small hoods are calculated so that they will exhaust the proper amounts of air and air supply to the rooms is calculated on the same basis. In every case this results in an air supply in excess of 30 cubic feet per minute per person. The air change in laboratories is more than twelve per hour. Furnished rooms are ventilated by duplex systems and concealed duct work. In all laboratories, all pipe and ventilation ducts are exposed. The duct work is finished with acid-resisting paint.

Special apparatus, such as incinerators, destructors, de-greasers, and similar apparatus used in research are treated with special exhaust systems. The Kjeldahl room and all chemical laboratories with steam baths have hood exhaust systems with remote control. One or more separate fans serve each laboratory or separate small groups of hoods on ceramic pipe lines and ceramic exhausters.

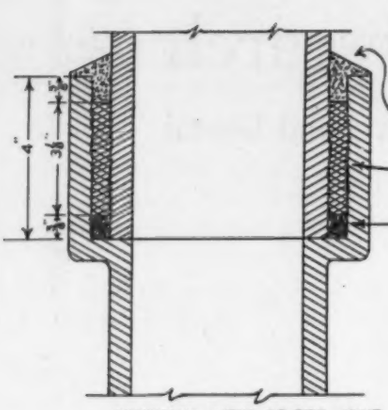
Ceramic pipe eight feet in length with special long-socket joints are used. The exhaust points of all ceramic fans at and above the roof are treated by providing special stone and burnt salt-glaze exhaust heads so designed that no fumes come in contact with any type of metal flashing. All hoods containing steam-making or fume-producing apparatus are built of specially constructed one-inch thick material.

Air Experimentation Room

The mechanical set up of physiological experimentation rooms for the study of individual thermal adjustment to changing conditions in the air environment is of particular interest to heating and ventilation engineers. Full details of this installation have not been completed, but the specifications require equipment capable of maintaining a test room temperature of 95 F in zero weather with steam at 5 pounds pressure and of cooling the room to 30 F in summer with 95 deg. dry bulb and 75 deg. wet bulb outside. Controlled relative



MEDICAL AND DENTAL RESEARCH BUILDING OF THE UNIVERSITY OF ILLINOIS. GRANGER & BOLLENBACHER, ARCHITECTS


5 PARTS BY WEIGHT OF TALC
1 PART BY WEIGHT OF ASBESTOS
MIXED WITH SILICATE OF SODA TO THE CONSISTENCY OF PUTTY.
ASPHALT MIXTURE SET HOT
ASBESTOS ROPE
SECTIONAL VIEW OF PIPE JOINT
DETAIL OF CERAMIC PIPE CAULKING

humidity conditions will range between 20 and 90 per cent at 95 F within the test room.

Wind velocity rates will be subject to control by means of portable fans operated independently by the research men in charge. Automatic recording instruments are not used, but the necessary precision thermometers and other instruments will be read by the research workers.

These test rooms are expected to carry forward the investigations of Dr. Lloyd Arnold into man's powers of resistance to contagious disease as a function of the ability to adjust to physical conditions in the air environment. The skin is the thermostat of the human body. Nervous adjustments set up through skin reactions to heat and cold, to moving air, to moist or dry conditions, to fluctuating atmospheres are basic to individual resistance to disease.

Two rooms will be devoted to this work, one for workers and apparatus, and the other for maintaining the climatic conditions it is desired to study. The work so far has been confined to animals, but the provision now contemplated can care for human patients when the entire matter has been sufficiently worked out for application to therapeutic uses. (*From an interview with Ernest V. Lippe*)—*S. P. M.*

Effect of Dust

It was not so long ago that, to the layman as well as to the physician and the engineer, dust consisted merely of specks which one might see in a sun-beam, worthy of attention only because it discolored our hands, faces and furnishings. Then we became gradually conscious of various other factors of dust, the extreme fineness of its particles, its penetration through microscopic passages or cracks, its abrasive action on machinery, its influence in respiratory diseases, etc. We became conscious of dust as an economic problem and as a carrier of bacteria.

Recently, however, we have been learning more about dusts that are harmful to human beings, aside from the bacteria the particles may carry.

In this connection it is interesting to note an article in the *Journal of Allergy* for November, 1930, written by Bret Ratner, M. D., of the department of pediatrics and immunology, University and Bellevue Hospital Medical College, New York University. Dr. Ratner states: "We must recognize the possibility of the development of sensitization to various dusts encountered in the industries, as well as in homes, schools, etc.

"There is growing evidence that contact with antigenic dusts may establish hypersentitiveness in susceptible individuals and that this should receive greater recognition in air conditioning." (Antigenic: A substance, the introduction of which into the body gives rise to the production of antibodies.)

In his conclusion, Dr. Ratner says: "The striking fact which is apparent from a comparison of cases cited from the literature and our experimental work is, therefore, that the same method of exposure in animal and man is capable of producing similar results, namely, that the continuous inhalation of such a dust as castor bean is capable of producing both toxic manifestations and anaphylactic sensitization [excessive susceptibility]. In other words, continued contact with an antigenic dust will cause the development of a sensitization to that specific dust in the case of susceptible individuals.

"We are not prepared to state what renders an individual susceptible. Many hypotheses have been offered but none so far has been proved adequate. We must grant, however, that there exists such a susceptibility, since not all individuals, not all animals, under similar circumstances, develop hyper-sensitiveness. Such being the case, we must recognize the possibility of the development of sensitization to various dusts encountered in the industries, as well as in homes, schools, etc. Until the basic cause for such susceptibility is discovered, in which event the individual may be able to cope with proteins, the only hope for any preventive measures in respiratory allergic conditions is to eliminate antigenic dusts from the environment, or possibly some manner may be discovered to render such dusts nonanaphylactogenic.

"The literature on castor bean sensitization offers an interesting situation for the study of this problem. Reports have been made on the development of such hypersensitiveness in isolated individuals working particularly in botanical laboratories. The occurrence of endemic asthma (Endemic: Peculiar to a district or locality, or a class of persons) in a community, due to this dust, has been described. The situation was duplicated experimentally in the guinea pig. The correlation of the findings in these situations, we believe, tends to support our original hypothesis that an antigenic dust in a dry state can act as a sensitizing agent, entry into the body being gained through the nasal mucosa. We, therefore, believe that dusts produced in industries, etc., may be significant factors in the production of certain respiratory allergic manifestations. We have suggested that the study of the dust problem may be facilitated by a method we have described, and that preventive measures against the inception of sensitization to specific antigenic dusts may eventuate from such a study."

Medical research is continually pointing out the harmful effects of dust, and consequently the need for its elimination from, or reduction in, outdoor air, and all types of buildings.

One of the most important developments of the many projects under consideration by the committee on steel of the American Society for Testing Materials at a recent meeting was the progress that has been made in the development of specifications for various types of fabricated pipe. Through the subcommittee on steel tubing and pipe there have now been prepared specifications for riveted pipe, lock-bar pipe, hammer welded pipe, electric resistance welded pipe, and electric fusion welded pipe.

Heating·Piping
and Air Conditioning

1895

American Society of Heating and Ventilating Engineers

JOURNAL SECTION

JUNE 1931

Material in this Section prepared by Publication Committee: W. A. Rowe, *Chairman;* E. O. Eastwood, and F. D. Mensing : A. V. Hutchinson, Manager of Publications

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Heat and Moisture Losses from Men at Work and Application to Air Conditioning Problems

By F. C. Houghten[1] (*MEMBER*), W. W. Teague[2] (*MEMBER*), W. E. Miller[3] (*NON-MEMBER*) and W. P. Yant[4] (*NON-MEMBER*), Pittsburgh, Pa.

HEAT and moisture dissipation from the human body, for various degrees of physical exertion and for a wide range of atmospheric conditions, has been the subject of investigation at the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS for the past four years. Reports giving the relation of the temperature, humidity, and air motion of the atmosphere to the rate of heat and moisture dissipation from the bodies of normally clothed men seated at rest were published two years ago[a-b]. The Laboratory has since published a paper[c] emphasizing the physiological and hygienic aspects of heat and moisture dissipation from the bodies of men working in still air, which paper deals with the practical application of these data to air conditioning problems. The findings of the Laboratory for men at rest found immediate application to the air conditioning and cooling of theaters, auditoriums, and other places of assemblage where large numbers of persons are seated at rest.

The results outlined herewith may find similar application to the conditioning of air in dance halls and other places of assemblage where people are active, and in industrial plants where the highest efficiency and the comfort of workers are desirable.

Practical Data and Their Application to Air Conditioning Problems

In the solution of problems involving air conditioning of space occupied by persons working or otherwise engaged in physical exertion the engineer is interested in the total rate of heat dissipation to the atmosphere by the average person. He is also interested in the rate of sensible and latent heat dissipation for all atmospheric conditions met within practice and the rate at which moisture is added to the atmosphere. In the solution of certain problems he is interested in knowing the percentage of the total loss which is dissipated as sensible and latent heat. These data for men working at three rates are given in convenient form for practical application in Figs. 1, 2, 3 and 4. Similar data for men at rest are also given for comparison.

Although total heat loss and sensible and latent heat

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[a, b, c] See Bibliography.

For presentation at the Semi-annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Swampscott, Mass., June, 1931.

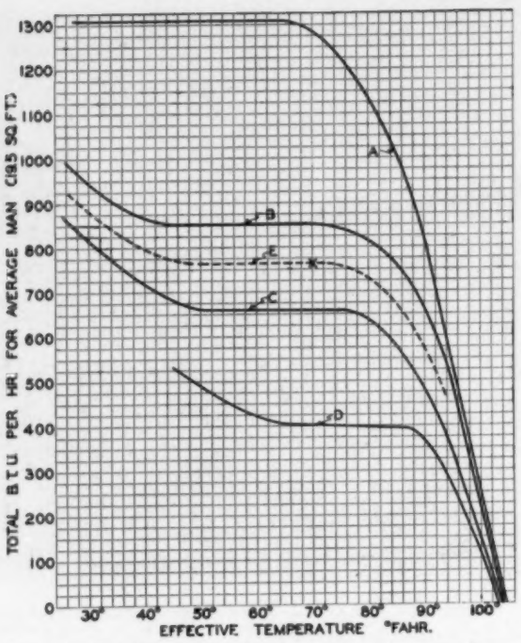

FIG. 1—RELATION BETWEEN TOTAL ENERGY LOSS FROM THE HUMAN BODY AND EFFECTIVE TEMPERATURE FOR STILL AIR*

* Curve *A*—Men working, 66,160 ft-lb per hour. Curve *B*—Men working 33,075 ft-lb per hour. Curve *C*—Men working, 16,538 ft-lb per hour. Curve *D*—Men seated at rest. Curves *A* and *C* drawn from data at an effective temperature of 70 deg. only and extrapolating the relation between curves *B* and *D*, which were drawn from data at many temperatures.

losses are not exact functions of effective temperature and dry-bulb temperature respectively for all conditions of humidity, they are plotted as such in the curves Figs. 1 to 4. This is accomplished by approximations not always rigidly accurate but sufficiently so for application

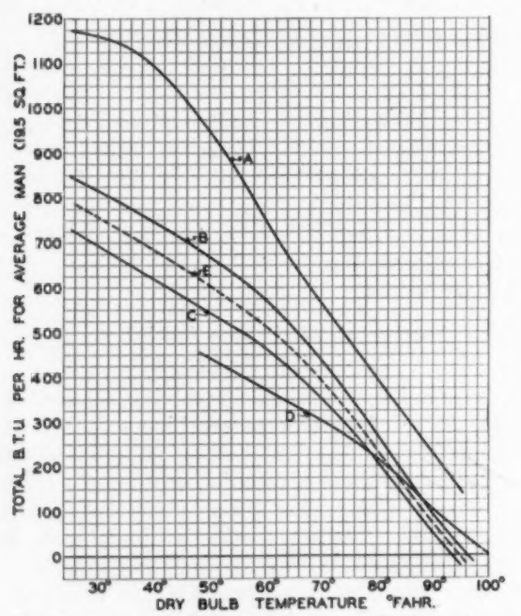

FIG. 2—RELATION BETWEEN SENSIBLE ENERGY LOSS FROM THE HUMAN BODY AND DRY-BULB TEMPERATURE FOR STILL AIR*

* Curve *A*—Men working, 66,150 ft-lb per hour. Curve *B*—Men working, 33,075 ft-lb per hour. Curve *C*—Men working, 16,538 ft-lb per hour. Curve *D*—Men seated at rest. Curves *A* and *C* drawn from data at a dry-bulb temperature of 81.3 F only and extrapolating the relation between curves *B* and *D* which were drawn from data at many temperatures.

to most practical problems. As will be shown later by the curves in Figs. 14 and 15, the latent and sensible heat losses show some variation with change in humidity for dry-bulb temperatures above 75 F.

The curves for men working at 33,075 ft-lb per hour were established by a large number of points at various temperatures and two humidities, this being the rate of work engaged in for the majority of the tests. The curves for work rates of 16,538 and 66,150 ft-lb per hour respectively were drawn from data on four men at an effective temperature of 70 F only and the application of the established relationship between the curves for men seated at rest and working at 33,075 ft-lb per hour. Notwithstanding these approximations and interpolations, the data in Figs. 1 to 4 are sufficiently accurate for most practical application. Where extreme accuracy is desired, particularly for definite humidities at high temperatures, it is recommended that the variation in the data for different atmospheric conditions as given in Figs. 11 to 17 be taken into consideration.

TABLE 1—RELATION BETWEEN METABOLIC RATE AND ACTIVITY ACCORDING TO VARIOUS AUTHORITIES

Activity	Metabolic Rate: Cal per Sq M per Hour	Metabolic Rate: Btu Per Hour for Avg. Man (19.5 Sq Ft Surface Area)	Authority
Seated at rest	53.4	384	*A*
Standing at rest	60	431	*A*
Going through motions necessary to operate work machine 7.5 times per minute	98	704	*A*
Operating work machine 7.5 times per minute, raising weight only	145	1042	*A*
Operating work machine 7.5 times per minute	118	848	*A*
Operating work machine 3.75 times per minute	96	690	*A*
Operating work machine 15 times per minute	180	1293	*A*
Walking 2 mph	106	761	*B*
Walking 3 mph	146	1049	*C*
Walking 4 mph	193	1388	*B*
Walking 4½ mph	241	1731	*C*
Walking 5 mph	352	2530	*C*
Slow run	318	2285	*D*
Maximum exertion (without food)	464-663+	3333-4762+	*D*
Level walking 2.2 mph	111	797	*E*
10 per cent grade 2.4 mph	233	1675	*E*
25 per cent grade 2.0 mph	414	2975	*E*
40 per cent grade 2.4 mph	559	4018	*E*
Very severe exercise	356	2555	*F*
Exercising more or less severely	85	610	*G*
Tailor	67	482	*H*
Bookbinder	87	626	*H*
Shoemaker	92	661	*H*
Carpenter	106-134	762-963	*H*
Metal worker	120	862	*H*
Painter (of furniture)	122	876	*H*
Stonemason	207	1488	*H*
Man sawing wood	250	1797	*H*

Authorities

A Research Laboratory, AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.
B Average values from (Douglas, Haldane, Henderson, and Schneider) and (Henderson and Haggard).
C Douglas, Haldane, Henderson and Schneider.
D Henderson and Haggard.
E H. M. Smith.
F Benedict and Carpenter.
G Atwater and Benedict.
H Becker and Hamalainen.

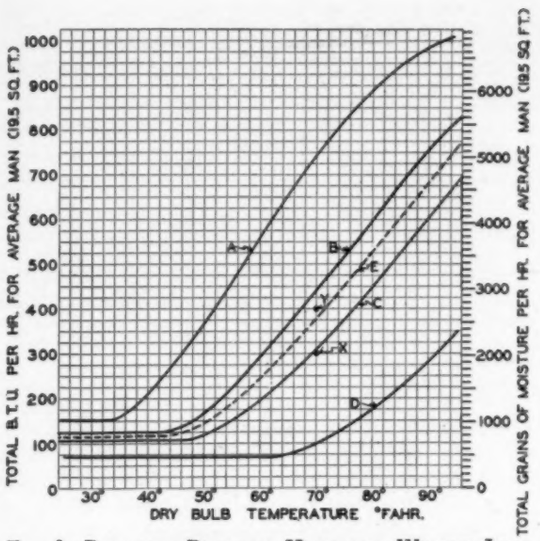

FIG. 3—RELATION BETWEEN HEAT AND WEIGHT LOSS FROM THE HUMAN BODY BY EVAPORATION AND DRY-BULB TEMPERATURE FOR STILL AIR*

* Curve *A*—Men working, 66,150 ft-lb per hour. Curve *B*—Men working 33,075 ft-lb per hour. Curve *C*—Men working, 16,538 ft-lb per hour. Curve *D*—Men seated at rest. Curves *A* and *C* drawn from data at a dry-bulb temperature of 81.3 F only and extrapolating the relation between curves *B* and *D* which were drawn from data at many temperatures.

Figs. 1 to 4 may be used in the solution of problems involving other types and rates of physical exertion than those for which the curves are drawn, provided the total rate of energy transformation within the body or the metabolic rate, and the rate at which energy is expended in performing external work which is not transformed into heat is known. Table 1 gives the metabolic rates for various degrees of activity. The metabolic rate for other types of activity may be estimated with the aid of the values in this table or they may be determined by independent metabolism tests.

By plotting the metabolic rate (or this rate less the rate of expending energy in external work) as point *x*, against the proper effective temperature in Fig. 1, a curve *E* may be drawn through this point similar to curves *A*, *B*, *C* and *D*. Curve *E* will then give the relation between total heat loss and effective temperature for this new rate of work. Likewise, curve *E* may be drawn in Figs. 2 and 3 bearing the same relation to curves *A*, *B*, *C* and *D* in these figures as the curve *E* does to these respective curves in Fig. 1.

In walking and many other types of physical exertion practically all the energy shown by metabolic tests, is transformed into heat within the body and external work need not be considered. As will be shown later in Fig. 10, the human body at best is extremely inefficient as an engine for performing external mechanical work and as a result the greater part of the energy transformation within the body as determined by metabolism becomes heat which must be dissipated to the air in either the sensible or latent form. For the type of work engaged in for this study, 11 per cent of the energy shown by metabolism left the body for external work. It is probable that few types of work will show a greater mechanical efficiency. Hence for most purposes rate of total heat dissipation may be taken as equal to the metabolic rate. If a closer approximation is desired, the efficiency of the body in performing such work must be estimated or determined by test.

Examples in the Use of Heat and Moisture Loss Data

Typical applications of the data given in Figs. 1 to 4 and in Table 1 are shown in the solution of the following problems:

Problem 1a. How much sensible heat, how much latent heat and how much water vapor will be added to the atmosphere per hour by 1,000 men while working at the rate of 33,075 ft-lb per hour in an atmosphere of 50 F dry-bulb and 45.5 F wet-bulb, respectively.

b. If the dry and wet-bulb temperatures were 75 and 59 F, respectively, how much heat and moisture would be dissipated to the atmosphere?

c. How do the losses in *(b)* compare with the same losses for 1,000 men seated at rest?

Solution—Problem 1a: From Fig. 2, curve *B*, find the sensible heat loss per person for a dry-bulb temperature of 50 F to be 670 Btu per hour. From Fig. 3 find the latent heat loss per person for a dry-bulb temperature of 50 F to be 165 Btu per hour and the moisture added to be 1,150 grains per hour. $1,000 \times 670 = 670,000$ Btu sensible heat; $1,000 \times 165 = 165,000$ Btu latent heat, and $1,000 \times 1,150 = 1,150,000$ grains, or 164 lb of water per hour will be added.

Solution—Problem 1b: From Figs. 2 and 3, respectively, curve *B*, the sensible and latent heat losses per person for a dry-bulb temperature of 75 F are found to be 360 and 515 Btu per hour. The water vapor added to the atmosphere is 3,480 grains per hour. The working men will then add 360,000 Btu sensible heat, 515,000 Btu latent heat and 3,480,000 grains or 497 lb of water vapor to the air per hour.

Solution—Problem 1c: From Figs. 2 and 3, respectively, curve *D*, find the sensible and latent heat losses at a dry-bulb tempera-

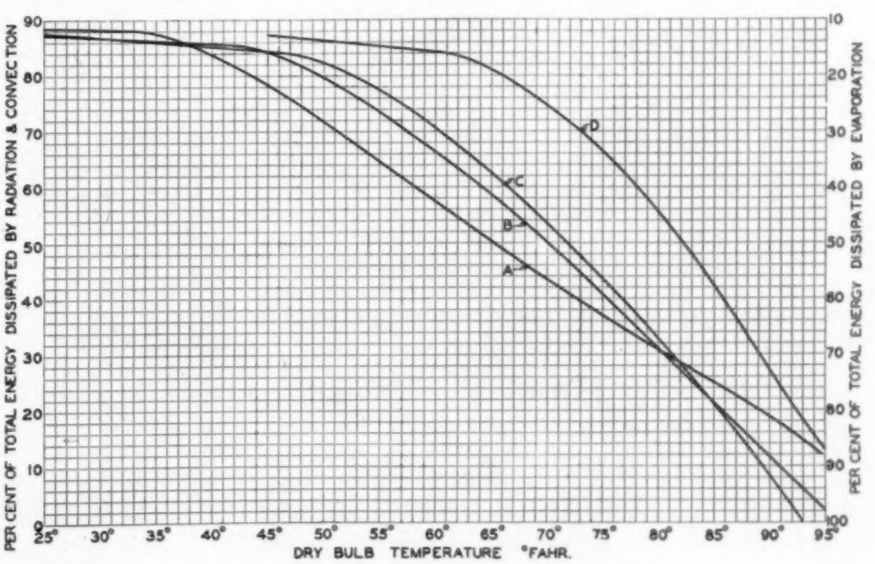

FIG. 4—RELATION BETWEEN ENERGY LOSS FROM THE HUMAN BODY BY EVAPORATION, RADIATION AND CONVECTION, AND DRY-BULB TEMPERATURE FOR STILL AIR*

* Curve *A*—Men working, 66,150 ft-lb per hour. Curve *B*—Men working, 33,075 ft-lb per hour. Curve *C*—Men working, 16,538 ft-lb per hour. Curve *D*—Men seated at rest. Curves *A* and *C* drawn from data at a dry-bulb temperature of 81.3 F only and extrapolating the relation between curves *B* and *D* which were drawn from data at many temperatures.

ture of 75 F for a person seated at rest to be 265 and 135 Btu per hour. The water vapor added to the air will be 900 grains per hour. The men will then add to the atmosphere 265,000 Btu sensible heat, 135,000 Btu latent heat and 900,000 grains or 129 lb of water vapor.

From Solution *(b)*, 360,000 — 265,000 = 95,000 Btu more sensible heat for men working at 33,075 ft-lb per hour than for men seated at rest. Hence, 515,000 — 135,000 = 380,000 Btu per hour more latent heat. Therefore, 497 — 129 = 368 lb per hour more water vapor.

Problem 2. Neglecting the gain or loss of heat to a building by transmission or infiltration through the walls, windows and doors, how many cubic feet of outside air with dry-bulb, wet-bulb and effective temperatures of 65 F, 59 F and 62.8 F, respectively, must be added per hour to a building containing 1,000 persons working at the rate of 33,075 ft-lb per hour in order that the inside dry and wet-bulb temperatures shall not exceed 75 and 65 F, respectively?

Solution—Problem 2: Figs. 2 and 3 give 360 Btu sensible heat and 3,480 grains of moisture as the additions per person with dry-bulb temperatures of 70 F in the building. Therefore, 360,000 Btu of sensible heat and 3,480,000 grains of moisture will be added to the air per hour.

Taking 0.24 as the specific heat of air, 2.4 Btu per pound of air will be required to raise the dry-bulb temperature from 65 to 75 F and $\frac{360,000}{2.4} = 150,000$ lb of air or $150,000 \times 13.4 =$ 2,010,000 cu ft of air per hour will be required. This is equivalent to $\frac{2,010,000}{60 \times 1,000}$ or 33.5 cu ft per person per minute.

The moisture content of the inside air as taken from a psychrometric chart is 76 grains per pound of dry air and that of the outside condition 65 grains, so the increase in moisture content will be 11 grains per pound of dry air; $\frac{3,480,000}{11}$ or 316,400 lb of dry air, or approximately 320,000 lb of air at the specified condition, will be required. This is equivalent to $320,000 \times 13.4 = 4,290,000$ cu ft of air per hour or $\frac{4,290,000}{60 \times 1,000} = 71.5$ cu ft of air per minute per person.

The higher volume of 71.5 cu ft per person per minute will be required to keep the wet-bulb temperature from rising above the 65 F specified. The dry-bulb temperature will, therefore, not rise to the maximum of 75 F.

Problem 3. A subway station has an average capacity of 5,000 persons. Assuming that the average state of activity of the crowd is walking 2 miles per hour, how much heat and moisture will be added to the atmosphere if the dry- and wet-bulb temperatures are maintained at 80 F and 66 F, respectively?

Solution—Problem 3: From Table 1 find the metabolic rate for walking 2 miles per hour to be 761 Btu per hour. Plotting this value in Fig. 1 at the constant portion of the curve and extrapolating the relation between curves *A*, *B*, *C* and *D*, dotted curve *E* is drawn for walking 2 miles per hour. Applying this same relationship in Figs. 2 and 3 gives curves for sensible and latent heat losses at the given state of activity.

From Figs. 2 and 3 find the sensible and latent heat losses per person at a dry-bulb temperature of 80 F to be 240 and 530 Btu per hour. The water vapor added will be 3,580 grains per hour. The occupants of the station will then add $5,000 \times 240$ or 1,200,000 Btu sensible heat, $5,000 \times 530$ or 2,650,000 Btu latent heat and $5,000 \times 3,580 = 17,900,000$ grains or 2,557 lb of water vapor to the air per hour.

Heat and Moisture Losses While Dancing

A graph of temperature of supply and exhaust air, and inside and outside relative humidity taken during the Junior Prom in Great Hall Memorial Union Building, University of Wisconsin, is shown in Fig. 5. One thousand persons are estimated to have danced at the

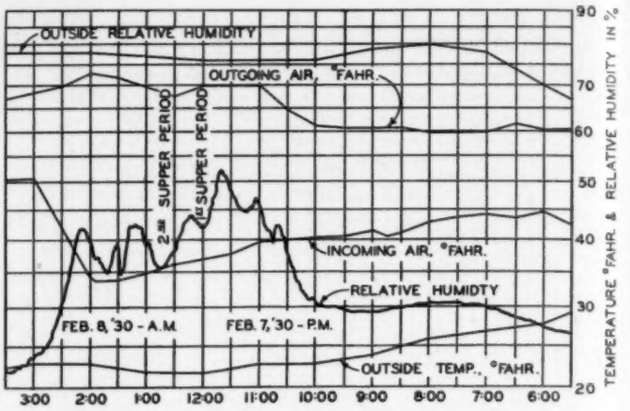
OUTSIDE RELATIVE HUMIDITY
OUTGOING AIR, °FAHR.
2nd SUPPER PERIOD
1st SUPPER PERIOD
INCOMING AIR, °FAHR.
RELATIVE HUMIDTY
FEB. 8, '30 - A.M.
FEB. 7, '30 - P.M.
OUTSIDE TEMP., °FAHR.
3:00 2:00 1:00 12:00 11:00 10:00 9:00 8:00 7:00 6:00
90 70 60 50 40 30 20
TEMPERATURE °FAHR. & RELATIVE HUMIDITY IN %

FIG. 5—TEMPERATURE AND HUMIDITY CHART FOR GREAT HALL THE MEMORIAL UNION BUILDING, UNIVERSITY OF WISCONSIN, FOR NIGHT OF FEB. 7-8, 1930

Junior Prom. This graph was supplied by Prof. G. L. Larson and is based on data collected by him in a study of ventilation requirements of the building and published by Prof. C. Braatz in The Wisconsin Engineer[d].

No heat or moisture was added to the atmosphere in the hall by the ventilating system during the dance. Hence the temperature and humidity rises are due to heat and moisture losses of those in attendance. The heating and humidifying effect of the dancers is clearly shown by the graph as pointed out by the authors of the university report as follows:

> The inside relative humidity curve indicates that the Grand March started about 9:45, at which time the relative humidity began to climb slowly for about twenty minutes, then dropped off momentarily during the picture taking ceremony, which was followed by a very decided rise during the first dance. From here on each of the following dances has a decided hump of its own on the relative humidity curve. Note also that while the temperature of the incoming air gradually decreased throughout the entire evening, the temperature of the air leaving the hall rose from about 61 F to 70 F during the first dance and remained practically constant for the remainder of the evening.

It is of interest to compare the calculated rates of latent heat and moisture addition to the hall as given by the authors of the university report, or calculated from their graph, with the values established in this report for persons working. In making these comparisons it should be emphasized that the values established by the Research Laboratory and here reported are for men and the heat and moisture units are given in English units for an average sized man or a 150-lb man 5 ft 8 in. in height, having a surface area of 19.5 sq ft. Perhaps a more correct average for women college students would be 120 lb in weight and 5 ft 4 in. in height which would give according to the chart by Du Bois[e-b] a surface area of 17 sq ft. This would give an average surface area for both men and women of 18.2 sq ft.

The authors calculate the moisture content of the outside or incoming air to be 12 grains per pound of dry air. Based upon an average relative humidity of 42 per cent at a dry-bulb temperature of 70 F for the period of the dance, the average moisture of the outgoing air was 46 grains per pound of dry air, or 34 grains were added per pound of dry air delivered to the hall. When the relative humidity was the highest point at 11:40 PM the moisture content was 57 grains, or 45 grains

[d] See bibliography.

were added per pound of dry air. The air delivery to the hall is given as 12,300 cfm or 738,000 cfh. The rate of moisture addition for the average relative humidity of 42 per cent and the highest relative humidity of 52 per cent is given by the equation

$$\text{Average rate} = \frac{738{,}000 \times 0.075 \times 34}{7{,}000} = 268.8 \text{ lb per hour.}$$

$$\text{Maximum rate} = \frac{738{,}000 \times 0.075 \times 45}{7{,}000} = 355.8 \text{ lb per hour.}$$

where 0.075 = weight per cubic foot of 70 F air.
and 7,000 = grains per pound.

The moisture addition per person is $\frac{268.8}{1000} = 0.2688$ and $\frac{355.8}{1000} = 0.3558$ lb per hour, and the latent heat added per person is $0.2688 \times 1036 = 278.5$ and $0.3558 \times 1036 = 368.6$ Btu per hour for the average and maximum rates, respectively. The rate of latent heat addition per hour per unit of surface area used in the charts in this report, or per 19.5 sq ft of body surface, is $278.6 \times \frac{19.5}{18.2} = 298.7$ Btu for the average time of the dance, and $368.5 \times \frac{19.5}{18.2} = 395.0$ Btu for the maximum rate. These two points are indicated by points x and y, respectively, Fig. 3. The maximum rate point y shows a rate of latent heat elimination slightly less than that for men working at the rate of 33,075 ft-lb per hour, or the average dancer probably expended energy at nearly as high a rate as he would while working at the rate of 33,075 ft-lb per hour on the Laboratory machine. By a similar comparison the average rate of energy dissipation by the dancers for the whole evening, including intermission and supper period, was about equal to that of a man working at the rate of 16,538 ft-lb per hour.

The abrupt rise and fall in the relative humidity curve with the dance activity compared with the relatively uniform dry-bulb temperature rise of the air bears out the findings of the Laboratory that sensible heat loss is little affected by activity or other factors for the same dry-bulb temperature and that the body controls latent heat dissipation so as to make the sum of sensible and latent heat loss equal to heat production in order to maintain the proper body temperature.

FIG. 6—BALANCE ON WHICH SUBJECTS ARE WEIGHED

Test Procedure and Observations

The results reported in this paper were obtained by making 135 heat balance tests on five subjects in various controlled atmospheric conditions in the psychrometric

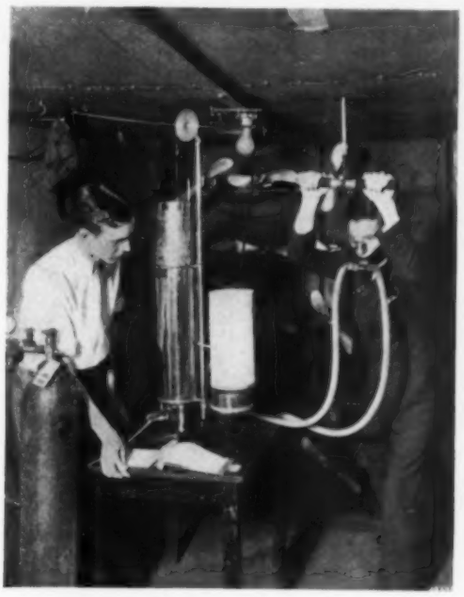

FIG. 7—BENEDICT-ROTH METABOLISM APPARATUS FOR DETERMINING METABOLIC RATE

chambers of the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in the Pittsburgh Station of the U. S. Bureau of Mines. These two rooms, in either one of which any desired atmospheric condition could be had and accurately controlled, are described in another Laboratory report[f]. The heat balance test procedure was in general similar to that published in an earlier Laboratory report[a-b]. Briefly, the subjects were made to work at a uniform rate in the desired atmospheric condition. Their weight loss and the consumption of O_2 were determined for the period of work, and from these observations the rate of oxidation of food, the rate of production and exhalation of CO_2, the rate of weight loss due to exhalation of carbon in the CO_2 produced and exhaled, the rate of weight loss due to evaporation of perspiration from the body surface and the respiratory tract and the rate of heat production in the body by metabolism were calculated. From these values and the change in body temperature, the total rate of heat dissipation to the atmosphere and its differentiation into sensible and latent heat losses were calculated.

Since energy production or metabolic rate varies to some extent with the diet of the subject the tests were made at various times during the day in reference to meals in order that the results might apply as nearly as possible to average working conditions.

In order that the results might be directly comparable with those reported[a-b] earlier for persons seated at rest without the introduction of other variables than the degree of activity, the clothing worn by the subjects was the same as that worn by the subjects in the earlier investigation. They were clothed in light athletic under-

[a,b,f] See bibliography.

wear, light-weight socks, low cut shoes, shirt with collar attached, and wool-cotton mixed business suits kept at the Laboratory for the purpose of such tests.

In reporting for the tests the subjects changed clothing, and then entered the air-conditioned room and

FIG. 8—WORK MACHINE

started working. After a 30-min preliminary work period the test was started and continued for one hour, excepting in tests at an effective temperature of 90 F when the test was shortened to 15-min preliminary and 30-min actual test periods because of the inability of the men to endure the high temperatures for a longer period.

Body temperatures (rectal) were taken at the beginning of the preliminary test period and at the beginning and end of the test period. These temperatures were needed in order to calculate the amount of heat storage in the body because of the heat capacity and change of temperature of the body during the test.

The subjects were weighed on the bullion balance (Fig. 6) to an accuracy of 0.2 gram at the beginning and end of the test period[a]. The weight loss was due to evaporation of moisture from the body surface and clothing and from the respiratory tract, and also due to the CO_2 exhaled less the O_2 consumed. The rate of O_2 consumption and CO_2 production were calculated from the O_2 consumption, as determined on a new model Benedict-Roth Recording Metabolism apparatus (Fig. 7), the operation of which is explained by Du Bois[e]. A spirometer is filled with pure O_2, from which the subject inhales through a mouth-piece, exhaling back into a closed system through a cannister of soda lime, which removes the CO_2 formed by the oxidation of food in the body. The drop of the spirometer bell indicates the volume of O_2 consumed and is recorded on a kymograph drum. The instrument is calibrated to read directly in calories, a unit of energy production within the body resulting from the oxidation of food, assuming an average respiratory quotient of 0.82, according to Benedict, Emmes, Roth and Smith[h]. This recorded value when corrected for the prevailing temperature of the O_2 in the spirometer and barometric pressure gave the rate of energy production or metabolic rate. There is evidence reported by Smith[i] that under some extreme conditions of work which required the same O_2 consumption as that reported in this paper, the respiratory quotient may be as high as 0.92. If

[a,e,h,i] See bibliography.

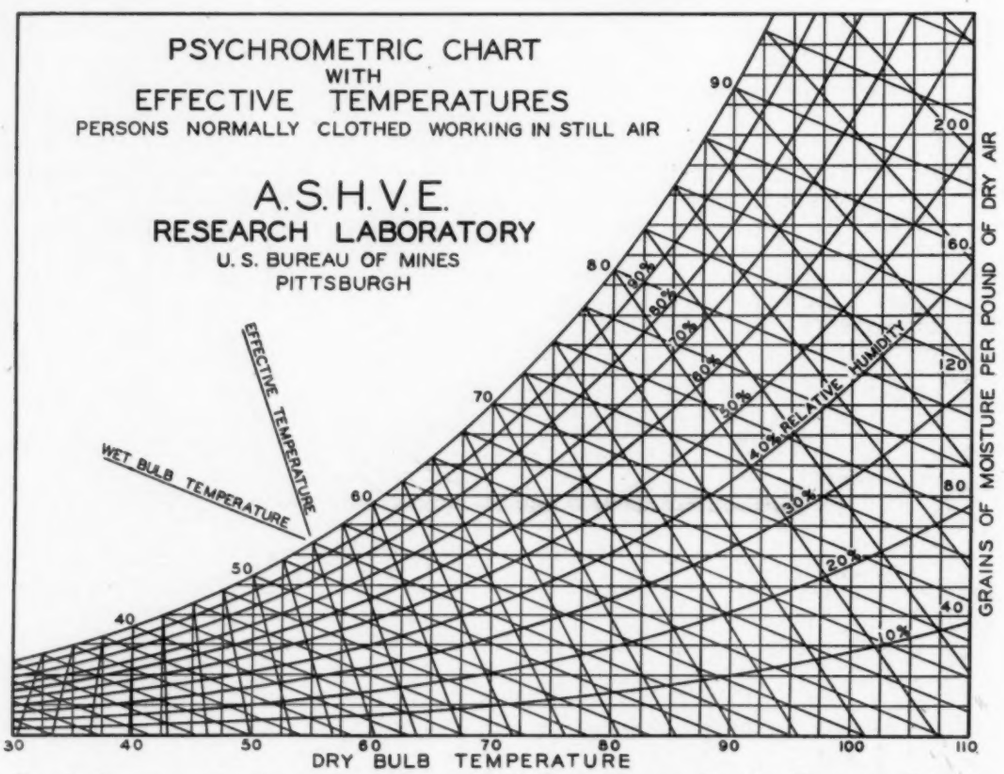

FIG. 9—PSYCHROMETRIC CHART WITH EFFECTIVE TEMPERATURES FOR PERSONS NORMALLY CLOTHED WORKING IN STILL AIR

this higher value had been used it would have increased the rate of heat production by 3 cal per square meter per hour, the heat loss by radiation and convection by 6 cal, and decreased the calculated heat loss by evaporation by 3 cal.

Total rate of heat loss, loss by radiation and convection combined and loss by evaporation were calculated as described in an earlier Laboratory report[a] from the metabolic rate, weight loss and body temperature rise. Briefly, the total energy loss was taken as the rate of energy production minus the rate of heat storage in the body due to its temperature rise and specific heat (0.83). The total heat loss was taken as the total energy loss minus the mechanical energy expended on the work machine. Heat loss by evaporation or the latent heat loss was computed from the weight loss by evaporation, and the heat of vaporization of water at body temperature. The radiation and convection, or sensible heat loss, was taken as the total heat loss minus the loss by evaporation. For convenience in the use of the instruments available the observations and most of the calculations were made in the metric system of units and the results were later translated into the English units for the convenience of the engineer.

The work was performed on a special work machine (Fig. 8), consisting of a 37-lb weight which was raised and lowered through a distance of 4 ft 10.8 in. by a rope over a pulley. In the majority of the tests, the weight was raised and lowered once every 8 seconds. The energy which was expended on the machine in raising the weight was taken as the product of the force necessary to raise the weight at the customary rate (44.5 lb), the distance through which it was raised (4 ft 10.8 in.), and the heat equivalent of one foot-pound of work (0.001285 Btu). The energy expended on the machine in raising and lowering the weight was taken as the product of the force necessary to raise the weight minus the force necessary to retard the fall to the desired rate, the distance through which it was raised, and the heat equivalent of work.

The psychrometric chart (Fig. 9), with effective temperature lines for individuals normally clothed working at the rate of 33,075 ft-lb per hour in still air, was used in analyzing the data presented. This chart was developed at the Research Laboratory and is very similar to other charts published by the Laboratory. The effective temperature lines in this chart have a slightly greater slope than those for individuals at rest[j], but a slightly lesser slope than those for individuals lightly clothed, working[k].

[a,b,c,j,k] See bibliography.

The data collected are for effective temperatures ranging from 27 to 90 deg at relative humidities of approximately 20 and 95 per cent. The limits of the effective temperature range were chosen as the extremes at which individuals could work at the prescribed rate in the clothing worn, without excessive discomfort.

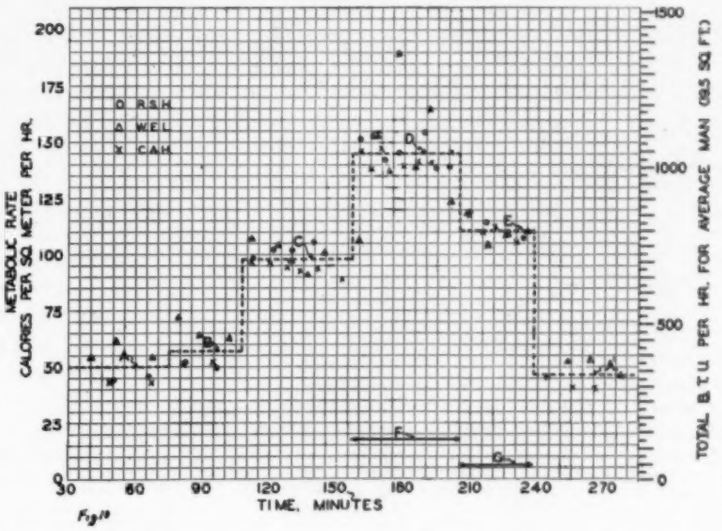

Fig. 10—Relation Between Metabolic Rate and Degree of Activity[a]

[a] *A* and *A'*—Seated at rest. *B*—Standing quietly. *C*—Body movement without raising and lowering weight. *D*—Raising weight only. *E*—Raising and lowering weight. *F*—Mechanical energy expended on machine in raising weight only. *G*—Mechanical energy expended on machine in raising and lowering weight.

Data and Results

The relation between metabolism and degree of physical activity is shown in Fig. 10. In this chart and those which follow, all temperatures are expressed in both degrees centigrade and degrees Fahrenheit. Heat loss values are expressed in calories per square meter of body surface per hour as is the accepted practice of the physiologist, and also in English units as Btu per hour for "a unit of surface area equal to that of an average sized man," that is, a 150-lb man, 5 ft 8 in. in height having a surface area of 19.5 sq ft or 1.81 sq m. Surface areas were determined from the chart by Du Bois[a,b].

Each of three subjects whose metabolic rates are distinguished by different symbols were seated quietly in a constant air condition of 70-deg effective temperature at 20 per cent relative humidity for the first 75 min of a 4¼ hour test period. Data were collected during the last 45 min of the rest period, showing an average metabolic rate *(A)*, Fig. 10, of 359 Btu per hour per average person. During the following 33 min they stood quietly and showed an average metabolic rate *(B)* of 409 Btu. During the next 50 min the subjects went through the customary movements of the body employed in raising and lowering the weight once every 8 seconds but without moving it and showed a metabolic rate *(C)* of 704 Btu. The following 48 min were occupied by the subjects in actually raising the weight once every 8 seconds but without lowering it, this being done by an attendant.

The metabolic rate *(D)* increased to 1042 Btu. During the following 33 min the subjects again went through the same body motions, this time actually raising and lowering the weight at the previous rate. Their metabolic rate *(E)* decreased to 790 Btu. During the last 45 min the subjects were again seated quietly and showed a metabolic rate *(A')* of 338 Btu per hour per average man.

The chart demonstrates how quickly metabolic pro-

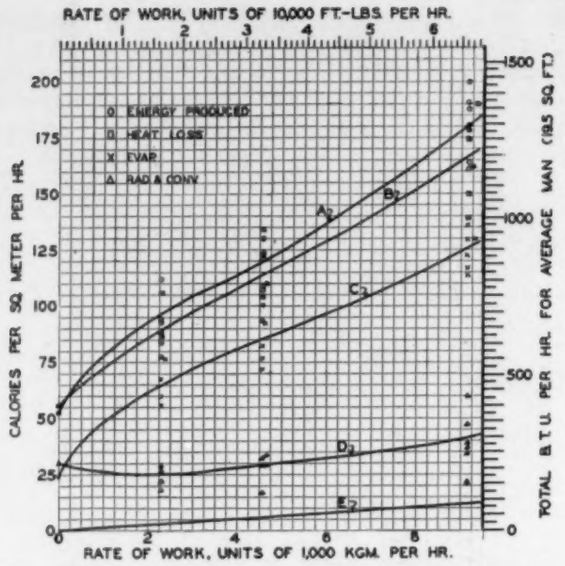
RATE OF WORK, UNITS OF 10,000 FT.-LBS. PER HR.
1 2 3 4 5 6
CALORIES PER SQ. METER PER HR.
TOTAL B.T.U. PER HR. FOR AVERAGE MAN (185 SQ. FT.)
O ENERGY PRODUCED
□ HEAT LOSS
× EVAP.
△ RAD & CONV.
A2
B2
C2
D2
E2
RATE OF WORK, UNITS OF 1,000 KGM. PER HR.

FIG. 11—RELATION BETWEEN ENERGY PRODUCTION AND HEAT LOSS FROM THE HUMAN BODY AND RATE OF WORK IN KILOGRAMS PER HOUR AND FOOT-POUNDS PER HOUR[a]

[a] Curve *A*—Energy production. Curve *B*—Total heat loss. Curve *C*—Heat loss by evaporation. Curve *D*—Heat loss by radiation and convection. Curve *E*—Mechanical energy expended on machine.

cesses, as measured by gaseous exchange, reach a new equilibrium when the degree of physical activity changes. The very low mechanical efficiency of the human body as a machine for performing this type of work is demonstrated by the chart. The mechanical energy expended on the machine to raise the weight, section *D*, or to raise and lower it, section *E*, is indicated by *F* and *G*, respectively. The mechanical energy is expressed in calories per hour per square meter of body surface area for the average subject and in Btu per hour for a body surface equal to that of an average sized man.

Raising and lowering the weight *E* requires the transformation of 381 Btu per hour of energy within the body more than is required when the subject is standing still, while the useful work performed *(G)* is only 42.1 Btu per hour. Assuming that the 409 Btu *(B)* are useful for maintaining life while standing and 381 Btu are needed for raising and lowering the weight, the human body as an engine for performing this type of work is only 11.1 per cent efficient. A like consideration showed that for raising the weight only *(F)* the body is 19.8 per cent efficient. The last figure is in close agreement with the value of 16 per cent reported by Murlin[1] as the result of laboratory experiments with the same type of work. It is also in agreement with the value of 20 per cent given by Yagloglou,[m] who plotted against the rate of work, the heat loss data, for men working, reported by Benedict and Cathcart[n] and Benedict and Carpenter.[o] On first thought it is surprising that more energy is required in raising the weight alone than is required for raising and lowering it. However, upon further consideration, the reason for this becomes apparent. While being lowered, the weight performs work in helping to raise the arms and body of the subject equal to its mass times the distance through which it falls, minus the work necessary to overcome friction in the system.

a.b.l.m.a.o. See bibliography.

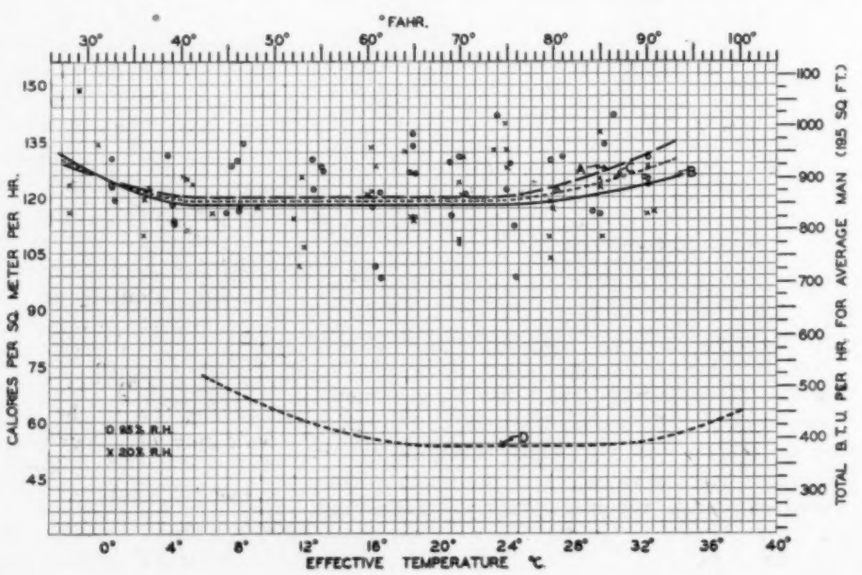
°FAHR.
30° 40° 50° 60° 70° 80° 90° 100°
CALORIES PER SQ. METER PER HR.
TOTAL B.T.U. PER HR. FOR AVERAGE MAN (185 SQ. FT.)
A
B
C
D 85% R.H.
× 20% R.H.
0° 4° 8° 12° 16° 20° 24° 28° 32° 36° 40°
EFFECTIVE TEMPERATURE °C.

FIG. 12 — RELATION BETWEEN ENERGY PRODUCTION IN THE HUMAN BODY AND EFFECTIVE TEMPERATURE*

* *A*—Energy production, at work, for 95 per cent relative humidity. *B*—Energy production, at work, for 20 per cent relative humidity. *C*—Average energy production, at work, all relative humidities. *D*—Average energy production, seated, at rest.

The relation between rate of working and energy production in the body, total heat loss, heat loss by evaporation, and heat loss by radiation and convection are given in curves *A, B, C* and *D,* respectively, Fig. 11. These data were all collected in a constant atmospheric condition of 70 deg effective temperature at 20 per cent relative humidity. The energy exchange at rest for these curves was taken from data published by the Research Laboratory[a-b] for persons at rest in the same air conditions. The three rates of work, 16,538, 33,075, 66,150 ft-lb per hour, were accomplished by raising and lowering the weight in the work machine at rates of once every 16, 8 and 4 seconds, respectively.

Curve *E* gives the rate at which mechanical energy was expended on the machine per square meter of the average body surface of the subjects and also for an average man. The relation between the metabolic rate or energy production and effective temperature for subjects raising and lowering the weight once every 8 seconds (working at the rate of 33,075 ft-lb per hour) is given in Fig. 12. The metabolic rate for individuals seated at rest is given in curve *D.*

The relation between effective temperature and total rate of energy and heat loss for subjects working at the rate of 33,075 ft-lb per hour is given in Fig. 13. The total rate of heat loss for subjects seated at rest and the rate at which mechanical energy was expended on the machine are given in curves *E* and *F,* respectively.

Latent and sensible heat losses from the entire body and from the respiratory tract are plotted against dry-bulb temperature in Figs. 14 and 15, respectively, for

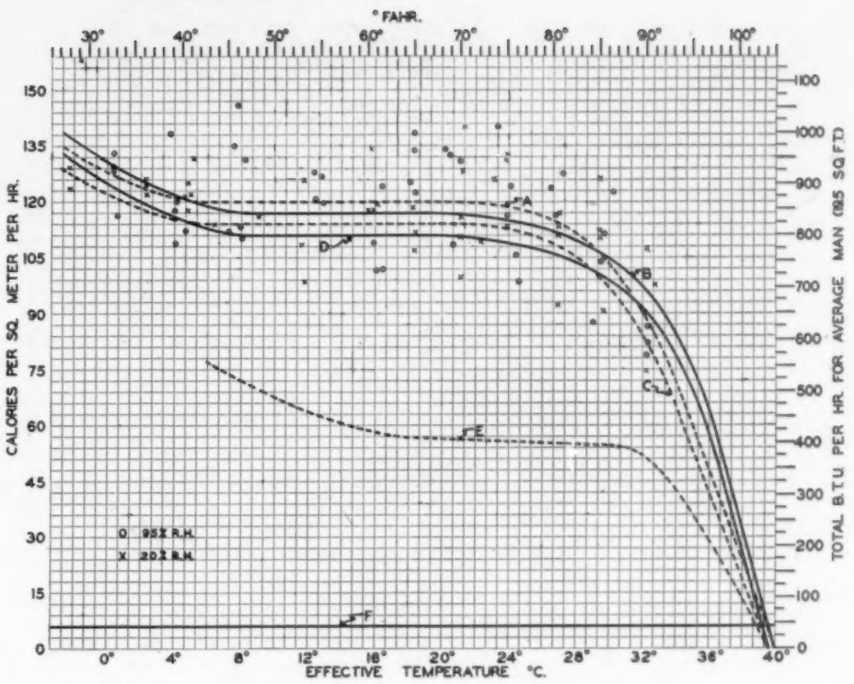

FIG. 13 — RELATION BETWEEN ENERGY AND HEAT LOSS FROM THE HUMAN BODY AND EFFECTIVE TEMPERATURE*

* *A*—Energy loss, at work, for 95 per cent relative humidity. *B*—Energy loss, at work, for 20 per cent relative humidity. *C*—Heat loss, at work, for 95 per cent relative humidity. *D*—Heat loss, at work, for 20 per cent relative humidity. *E*—Average heat loss, seated at rest. *F*—Mechanical energy expended on machine.

men working at the rate of 33,075 ft-lb per hour. The losses for men seated at rest are also given for comparison in the broken line curves. Heat and moisture losses from the respiratory tract of men working were calculated from the volume of normal air they would inhale and exhale for this rate of O_2 consumption, the temperature and moisture content of the air for each test condition and the temperature and degree of saturation of exhaled air as found by experiment.[a] Respiratory losses are shown to be very small in comparison to other losses.

The relation between effective temperature at 95 per cent relative humidity and total heat loss, heat loss by radiation and convection and heat loss by evaporation are given in Fig. 16 for subjects working at the rate of 33,075 ft-lb per hour. In Fig. 17 similar data are given for air conditions of 20 per cent relative humidity. The broken line curves give the losses for men seated at rest for comparison.

The curves showing thermal reactions of men at work in Figs. 11 to 17 are similar in shape to the corresponding curves for subjects at rest, as previously published by the Laboratory[a-b] and shown in these charts for comparison. The magnitude of most of the reactions are, however, much greater for men working. Any characteristic changes in the slope in the working curves occur at lower temperatures than in the corresponding curves for rest.

A minimum energy production of 855 Btu per hour (119 cal per square meter per hour) for an average man is indicated, which agrees with other investigators previously cited. Energy production and loss remain fairly constant over an effective temperature range of from about 43 to 75 deg and when plotted against effective temperature are practically independent of relative humidity, indicating that these reactions are functions of the effective temperature scale. If plotted against dry-bulb temperature, these reactions show considerable separating out for the two extreme humidities studied. Above an effective temperature of 75 deg, the total heat loss decreases rapidly until it reaches zero at about 103 F or at an effective temperature equal to 103 F body temperature. This decrease in heat loss, however, is not accompanied by a decrease in energy production but rather by an increase or in other words, the human body while working at this rate loses thermal control in an atmosphere above an effective temperature of 75 deg.

Fig. 11 shows that increased rate of work is accompanied by a rapid

a-b. See bibliography.

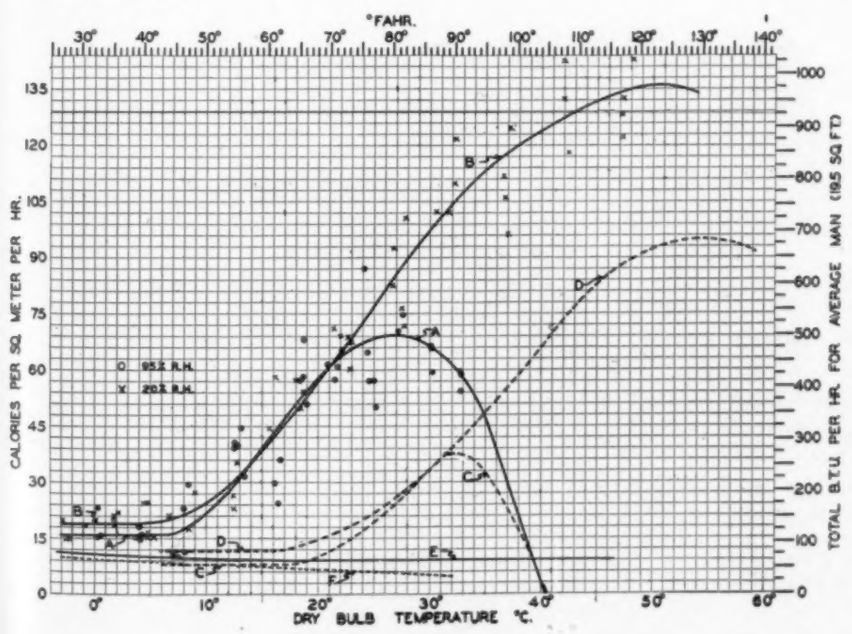

FIG. 14 — RELATION BETWEEN HEAT LOSS BY EVAPORATION FROM THE HUMAN BODY AND DRY-BULB TEMPERATURE*

* *A*—From body surface and respiratory tract, at work, for 95 per cent relative humidity. *B*—From body surface and respiratory tract, at work, for 20 per cent relative humidity. *C*—From body surface and respiratory tract, seated at rest, for 95 per cent relative humidity. *D*—From body surface and respiratory tract, seated at rest, for 20 per cent relative humidity. *E*—From respiratory tract, at work, at 20 per cent relative humidity. *F*—From respiratory tract, at work, at 95 per cent relative humidity.

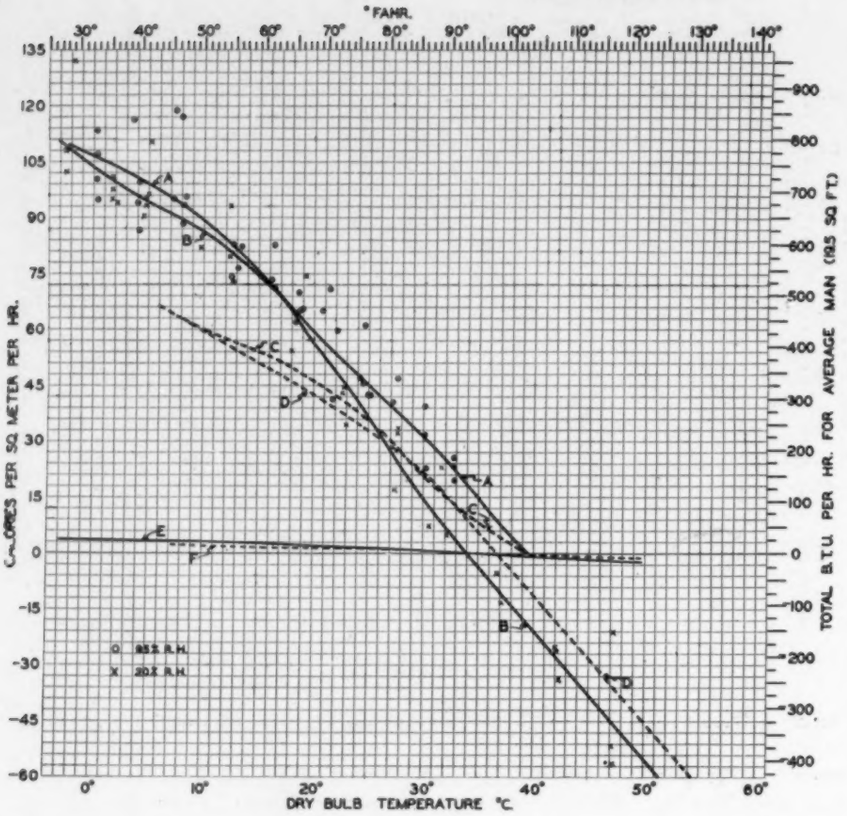

FIG. 15 — RELATION BETWEEN HEAT LOSS BY RADIATION AND CONVECTION FROM THE HUMAN BODY AND DRY-BULB TEMPERATURE*

* *A*—From body surface and respiratory tract, at work for 95 per cent relative humidity. *B*—From body surface and respiratory tract, at work, 20 per cent relative humidity. *C*—From body surface and respiratory tract, seated at rest, for 95 per cent relative humidity. *D*—From body surface and respiratory tract, seated at rest, for 20 per cent relative humidity. *E*—Sensible heat loss from respiratory tract, at work, for 20 per cent and 95 per cent relative humidity. *F*—Sensible heat loss from the respiratory tract seated at rest, for 20 per cent and 95 per cent relative humidity.

increase in energy transformation within the body many times greater than the rate of increase in the mechanical energy expended on the machine. Total heat loss and heat loss by evaporation also increase with work. Heat loss by radiation and convection, however, increases slowly. This is further borne out by the curves in Figs. 14 to 17. At a dry-bulb temperature of 68 F, the heat loss by radiation and convection for men working is only about 33 per cent greater than for men seated at rest. At the same temperature the heat loss by evaporation is 375 per cent greater for work than it is for resting conditions. This is in keeping with the physical conception of heat loss, for heat loss by radiation and convection should theoretically be independent of physiological control except for a slight variation in skin temperature. Physiological control of body temperature is obtained primarily by control of perspiration available for evaporation. The increase of heat loss by radiation and convection from the bodies of working subjects over the same loss for subjects at rest as indicated in Figs. 11 and 15 is probably due to increased ventilation of the body surface by the bellows effect of the clothes and the general air motion produced by the movement of the worker.

At dry-bulb temperatures of 103 F and 93.2 F, the heat loss by radiation and convection becomes zero for relative humidities of 95 and 20 per cent, respectively. The fact that heat loss by radiation and convection becomes zero at a lower temperature at 20 per cent relative humidity than it does at 95 per cent is probably accounted for by a lowered skin and clothing temperature at 20 per cent relative humidity due to evaporation of perspiration. The high temperature of 103 F, at which radiation and convection becomes zero at a relative humidity of 95 per cent, is probably fixed by the high body temperature and skin temperature attained by the subjects while working at this high temperature.

The composite charts (Figs. 16 and 17) illustrate the variation in thermal reactions with effective temperature at the same relative humidity. Total heat loss when plotted against effective temperature is independent of relative humidity. Radiation and convection and evaporation losses vary greatly with relative humidity for the

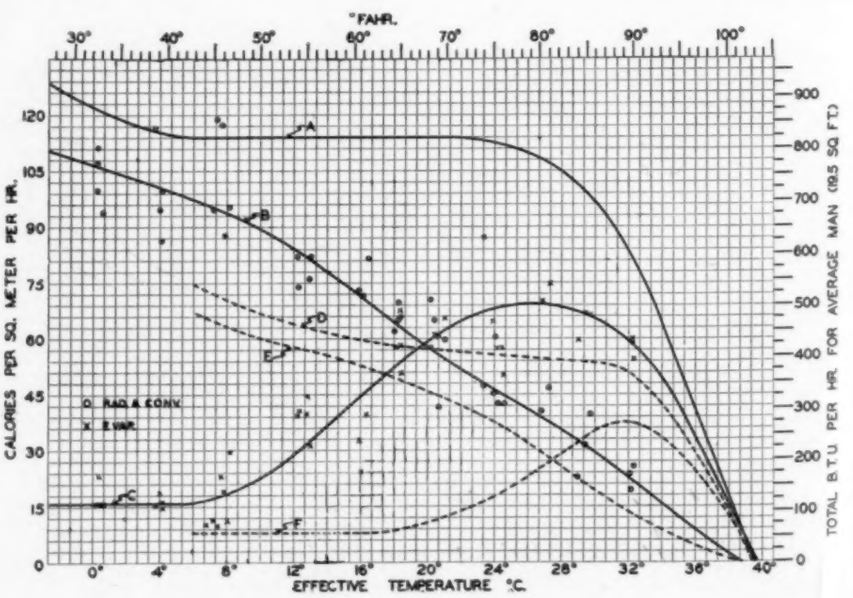

FIG. 16—RELATION BETWEEN HEAT LOSS FROM THE HUMAN BODY AND EFFECTIVE TEMPERATURE AT 95 PER CENT RELATIVE HUMIDITY*

* *A*—Total heat loss, at work. *B*—Loss by radiation and convection, at work. *C*—Loss by evaporation, at work. *D*—Total heat loss, seated at rest. *E*—Loss by radiation and convection, seated at rest. *F*—Loss by evaporation, seated at rest.

same effective temperature as indicated by a comparison of the curves in the two charts. At an effective temperature of 61.5 deg and a relative humidity of 20 per cent, radiation and convection and evaporation losses for subjects working are equal, each accounting for half of the entire heat loss, while at the same temperature at 95 per cent relative humidity, evaporation loss is only 39.7 per cent of the total, the other 60.3 per cent being accounted for by radiation and convection.

The relation between degree of comfort experienced by the subjects while working and the effective temperature of the atmosphere is given in Fig. 18. Each man expressed his feeling of warmth at equal intervals three times during each test at any atmospheric condition as too cold, too cool, comfortably cool, comfortable, comfortably warm, too warm and too hot. These expressions were arbitrarily given numerical values of from 1 to 7, respectively, The points plotted are the averages of the three expressions given by each man in a given test. Taking the comfort zone as that range of effective temperature between comfortably cool and comfortably warm or from numerical value 3 to 5, the comfort zone extends from 46 to 64 deg effective temperature. The comfort line for this degree of work is 53 deg effective temperature.

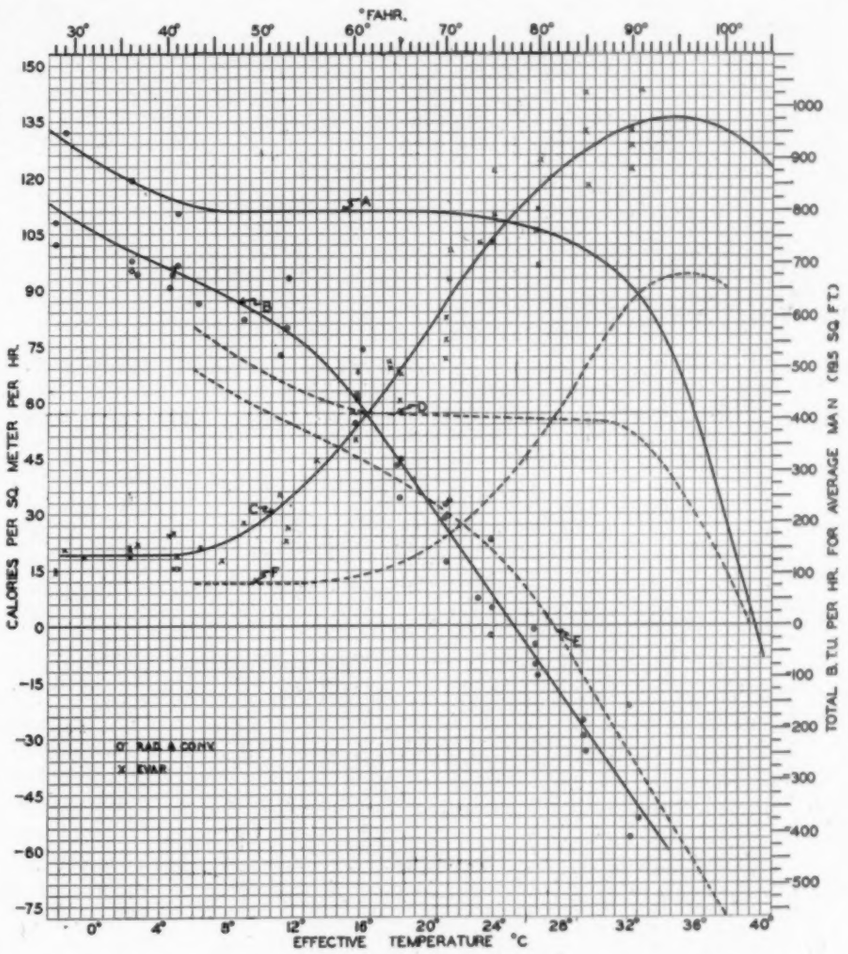

FIG. 17 — RELATION BETWEEN HEAT LOSS FROM THE HUMAN BODY AND EFFECTIVE TEMPERATURE AT 20 PER CENT RELATIVE HUMIDITY*

* *A*—Total heat loss, at work. *B*—Loss by radiation and convection, at work. *C*—Loss by evaporation, at work. *D*—Total heat loss, seated at rest. *E*—Loss by radiation and convection, seated at rest. *F*—Loss by evaporation, seated at rest.

The comfort line as located above is 11 deg lower than 64, as previously established by the Laboratory[p] for persons normally clothed at rest. The relation of the comfort line to the limits of the comfort zone as established for men working is similar to that reported earlier for men at rest, that is, the comfort line is nearer the lower limit of the zone than the upper limit. This means that a person is more sensitive to a few degrees of cooling below an ideally comfortable temperature than he is to the same degree of rise of temperature above. It should be emphasized that the comfort zone as established above is for men wearing coat and vest and working at the definite rate reported. In practice the coat and vest would be discarded when the man began to feel too warm and as a result he would be comfortable over a wider range of temperatures.

The effective temperature at which perspiration changes from insensible to sensible for men working is lower than for men at rest. This is indicated by the heat-loss-by-evaporation curves for persons at rest and working in Figs. 16 and 17. The atmospheric conditions at which various degrees of perspiration take place for men working are given in Table 2. The lowest degree for which 40 per cent of the subjects experienced the condition of perspiration indicated is given for both 20 and 95 per cent relative humidity. For comparison the conditions at which the same degree of perspiration takes place for persons at rest are also given.

[p] See bibliography.

Summary

1. This report contains data in the form of curves giving the rate of total heat loss, heat loss by radiation and convection and heat loss by evaporation for men working at three constant rates in still air at various temperatures and two humidities. Corresponding curves for these losses with subjects seated at rest are also given.

2. Data for practical application and examples showing how it may be used in the solution of problems in air conditioning are presented.

3. A table showing the metabolic rate for different kinds of activity is given and the application to practical problems is shown.

4. Total energy production, energy loss and heat loss are shown to be functions of effective temperature.

5. Sensible and latent heat loss are shown to be functions of dry-bulb temperature and only slightly affected by relative humidity except at extreme temperatures.

6. Sensible heat loss for men working increases but little over this loss for men at rest.

7. Latent heat loss increases rapidly with physical activity and is depended upon almost solely by the body for maintaining a constant body temperature with varying rates of heat production.

8. The comfort zone for men normally clothed and working at 33,075 ft-lb per hour is given as 46 to 64 deg effective temperature and the comfort line as 53 deg.

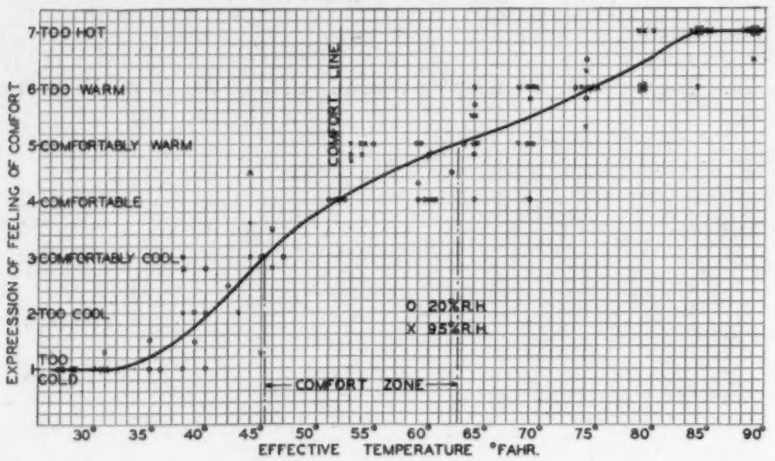

FIG. 18—RELATION BETWEEN COMFORT AND EFFECTIVE TEMPERATURE FOR MEN WORKING, 33,075 FT-LB PER HOUR

9. The degree of perspiration experienced by men working at 33,075 ft-lb per hour is given for various temperatures and humidities.

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TABLE 2—CONDITIONS OF SENSIBLE PERSPIRATION FOR VARIOUS ATMOSPHERIC CONDITIONS

Degree of Perspiration*	Atmospheric Condition											
	Men Working 33075 Ft.-Lb. per Hour						Men Seated at Rest					
	95 Per Cent Relative Humidity			20 Per Cent Relative Humidity			95 Per Cent Relative Humidity			20 Per Cent Relative Humidity		
	E.T.	D.B.	W.B.	E.T.	D.B.	W.B.	E.T.	D.B.	W.B.	E.T.	D.B.	W.B.
Forehead clammy	59.0	59.4	58.3	69.5	80.5	56.5	73.0	73.6	72.4	75.0	87.0	60.7
Body clammy	50.0	50.2	49.3	57.0	61.6	44.2	73.0	73.6	72.4	75.0	87.0	60.7
Body damp	60.0	60.3	59.3	62.5	69.6	49.5	79.0	79.7	78.4	81.0	97.5	67.5
Beads on forehead	68.0	68.5	67.5	76.0	91.0	63.4	80.0	80.8	79.4	87.0	109.4	75.2
Body wet	69.0	69.6	68.5	71.0	82.8	53.0	84.5	85.4	84.0	86.5	108.5	74.6
Perspiration on forehead runs and drips	78.5	79.3	78.0	82.0	100.5	70.2	88.0	89.0	87.6	94.0	125.2	85.4
Perspiration runs down body	79.0	79.8	78.5	81.0	99.8	69.0	88.5	89.5	88.1	90.0	116.0	79.5

*40 per cent of subjects registered degree of perspiration equal or greater than indicated.

Flow of Steam Through Orifices Into "Radiators"

By Sterling S. Sanford[1] (*MEMBER*) and Carl B. Sprenger[2] (*NON-MEMBER*), Detroit, Mich.

THE "radiation" installed in buildings must emit sufficient heat to maintain the desired temperature in the coldest weather. For higher outside temperatures, the "radiators" will be of greater capacity than required, and overheating will result unless the heat output is reduced. The amount of heat delivered by a steam "radiator" can be reduced by any one of the following three methods:

1. By lowering the pressure and thus the temperature of the steam.
2. By admitting steam to the "radiator" intermittently and thus reducing the length of time it is filled with steam.
3. By restricting the flow of steam into the "radiator."

In using the last method, the rate of flow of steam into a "radiator" can be controlled by means of (1) a restriction of variable area such as is obtained by throttling the radiator valve, or (2) a fixed restriction, such as a fixed orifice, a variable flow being obtained by varying the pressures on the two sides of the orifice. This paper deals with the last-named method of controlling heat output, that is, by the use of a fixed orifice in the "radiator" inlet.

The use of a fixed orifice makes it possible to reduce the heat output of a "radiator" to any desired extent. The "radiator" may be kept full of steam when the maximum heat output is needed, by adjusting the pressure on the inlet side of the orifice so that it is sufficiently above that in the "radiator." In mild weather the flow of steam into the "radiator" may be reduced to such an extent by reducing the difference in pressure between the two sides of the orifice that only a small part of the "radiator" is heated. Between these extremes, the pressure differential may be adjusted to give the flow of steam required for weather conditions encountered at any time during the heating season.

"Radiator" orifices are installed in various ways. Fig. 1 shows three methods of application. At *A*, a disc containing a straight-sided orifice is inserted in a recess cut in the "radiator" end of the valve tail-piece. The orifice discs of this type used in the tests described were 3/32 in. thick. The orifice at *B* is in a copper cup-shaped stamping 0.023 in. thick and is inserted in the union end of the tail-piece. The orifice at *C* is in a brass stamping 0.019 in. thick shaped to form a gasket in the union.

"Radiator" orifices have been used in steam heating systems for many years, yet so far as the authors know, no reliable data on the flow of steam through them have been published. Such data as are available are for the flow of steam through orifices installed in straight runs of piping and in general are for higher pressures or for larger orifice sizes than those used for "radiators." The tests described in this paper were made to determine the flow through different sizes of orifices under various pressure conditions for controlling temperature in buildings.

Description of Apparatus and Tests

The apparatus used in making the tests is shown in Figs. 2 and 3. The apparatus indicated in Fig. 2 is as follows:

A—A steam separator for removing entrained moisture from the steam. The separator was dripped through a special float trap.

B—A ¾-in. pressure reducing valve used for reducing the pressure of the district steam supply from 30 lb per square inch to 3 lb per square inch gage.

C—A ¾-in. globe valve used for throttling the steam to the desired pressure ahead of the orifice.

D—A glass-stem thermometer inserted in a well in the steam pipe.

E—A ½-in. thermostatic trap for dripping the steam pipe ahead of the radiator valve.

F—A bottle with tight connections for collecting the drip from the trap during sub-atmospheric pressure tests.

G—A mercury manometer giving the pressure ahead of the radiator valve.

H—A mercury manometer giving the pressure drop through the radiator valve and orifice.

I—A mercury manometer giving the pressure in the radiator.

J—A ¾-in. standard radiator valve.

K—The valve tail-piece in which the orifice was installed.

L—A 10-section, 2-column radiator, 32 in. high.

M—A galvanized iron tank filled with cooling water in which the radiator was immersed.

N—The cold water supply.

O—The overflow for the tank.

P—The bottle in which the condensate was collected and weighed.

Q—A motor-driven lab-

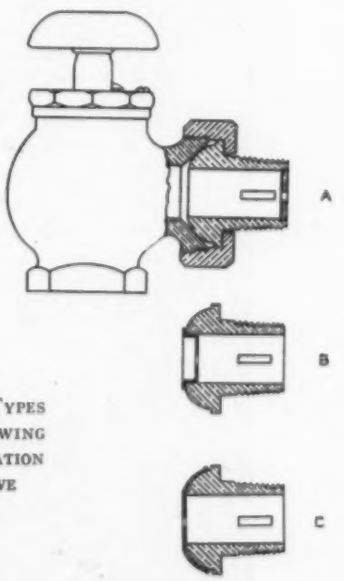

FIG. 1—DIFFERENT TYPES OF ORIFICES SHOWING METHOD OF INSTALLATION IN RADIATOR VALVE

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[2] Engineer, Central Heating Department, The Detroit Edison Co., Detroit, Mich.
For presentation at the Semi-annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Swampscott, Mass., June, 1931.

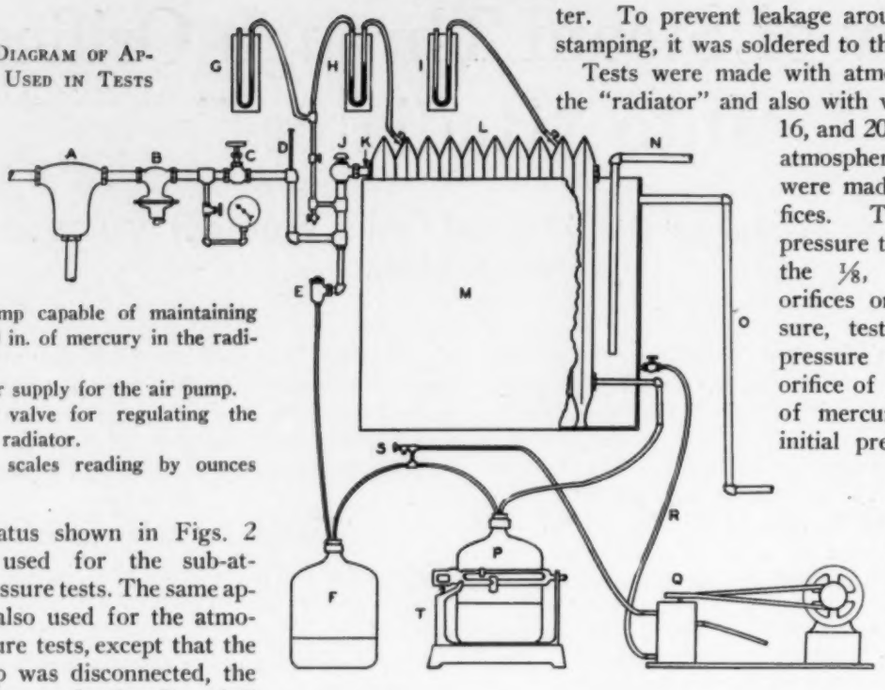

FIG. 2—DIAGRAM OF APPARATUS USED IN TESTS

oratory air pump capable of maintaining a vacuum of 20 in. of mercury in the radiator.

R—The water supply for the air pump.

S—A needle valve for regulating the vacuum in the radiator.

T—Platform scales reading by ounces to 50 lb.

The apparatus shown in Figs. 2 and 3 was used for the sub-atmospheric pressure tests. The same apparatus was also used for the atmospheric pressure tests, except that the vacuum pump was disconnected, the connection between bottles *F* and *P* eliminated, and the "radiator" and bottles *F* and *P* vented to the atmosphere.

The data presented in Figs. 4 to 10, inclusive, are from tests with the types of orifices designated as *A* and *B* in Fig. 1. The orifice plate marked *A* was a brass disc 3/32 in. thick and 13/16 in. in diameter. The orifices were drilled 1/64 in. undersize in a drill press and reamed by hand to the desired size. All burrs were removed. The diameters of the orifices tested were 3/32, 7/64, ⅛, 5/32, 11/64, and ¼ in. The plate marked *B* in Fig. 1 was a copper stamping 0.023 in. thick, with the orifice drilled and reamed as for *A*. Three sizes were tested, ⅛, 11/64, and ¼ in. in diameter. To prevent leakage around the edge of the stamping, it was soldered to the tail-piece.

Tests were made with atmospheric pressure in the "radiator" and also with vacuums of 4, 8, 12, 16, and 20 in. of mercury. The atmospheric-pressure tests were made on all of the orifices. The sub-atmospheric pressure tests were made with the ⅛, 11/64, and ¼-in. orifices only. At each pressure, tests were made with pressure drops through the orifice of ½, 1, 2, 4, and 6 in. of mercury by changing the initial pressure ahead of the radiator valve.

The flow of steam through the orifice was determined by weight. This was obtained by condensing the steam in a "radiator" surrounded by cold water and collecting the condensate in a bottle on platform scales. With few exceptions each test ran for one hour. During each test the pressures on the inlet side of the "radiator" valve and in the "radiator" were held as nearly constant as was possible. Any visible deviation from the test pressure was corrected immediately. During the tests at atmospheric pressure, the "radiator" pressure was kept constant by venting the "radiator" to the atmosphere. During the sub-atmospheric pressure tests, the vacuum in the "radiator" was held constant by manipulation of the needle valve *S*, varying the amount of air admitted to the air pump. The pressure ahead of the "radiator" valve was held constant by

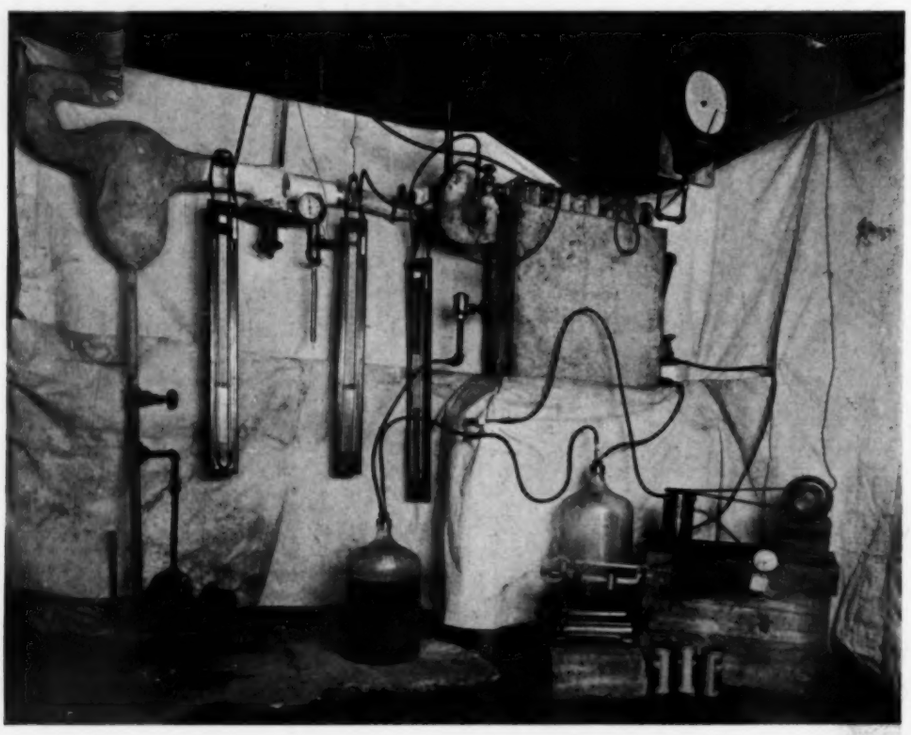

FIG. 3—TEST APPARATUS

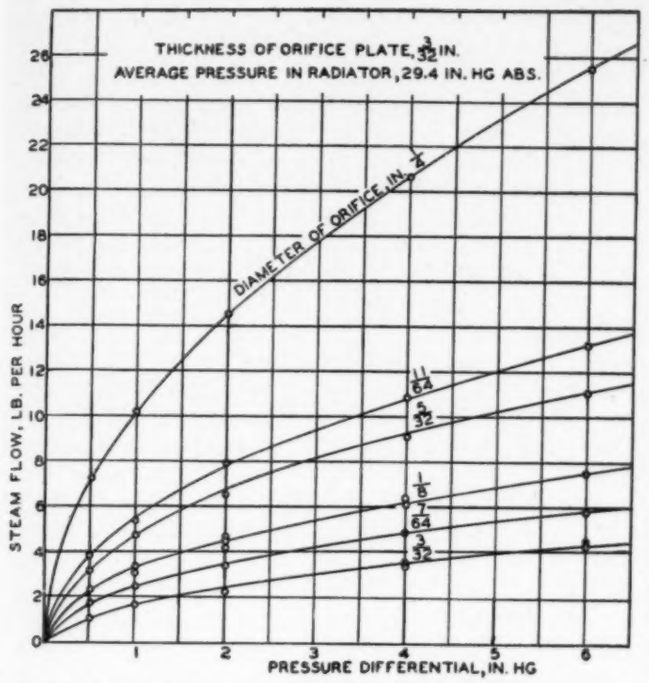
THICKNESS OF ORIFICE PLATE, 3/32 IN.
AVERAGE PRESSURE IN RADIATOR, 29.4 IN. HG ABS.
DIAMETER OF ORIFICE, IN.
STEAM FLOW, LB. PER HOUR
PRESSURE DIFFERENTIAL, IN. HG

FIG. 4—FLOW OF STEAM VS. PRESSURE DROP THROUGH ORIFICE FOR ORIFICES IN A PLATE 3/32 IN. THICK

manipulation of the globe valve *C*.

In starting a test, the pressures were held constant for about 15 minutes before taking the initial scale reading. This allowed the temperature and other conditions to reach equilibrium. Pressure and temperature readings were recorded every 10 minutes. The time required to accumulate equal weights of condensate in the bottle *P* was recorded as the test progressed. If the time varied appreciably indicating non-uniformity of flow, the test was continued until an hour of uniform flow was obtained. In checking the time, the weight was moved along the scale beam in increments corresponding to the condensation taking place in 10 min. The electric bell attached to the scales (Fig. 3) assisted in making this check.

No trap was used on the "radiator" discharge because the condensation rate of the water-cooled "radiator" was sufficient to prevent steam from reaching the outlet. During the sub-atmospheric pressure tests, bottles *F* and *P* were connected as shown so that the same vacuum existed in them as in the "radiator." The discharge of trap *E* was thus always at a lower pressure than its inlet.

Moisture in the steam was supposedly removed by the separator *A*, and any which remained was evaporated by the reduction in pressure from 30 to 3 lb per square inch in passing through the reducing valve *B*. The piping was well covered, the distance from the reducing valve to the orifice was small, and the piping was dripped through a trap ahead of the orifice. Nevertheless, it was feared that there might be moisture in the steam, and since the flow was being determined by weight, it was very important not to have any moisture in the steam passing through the orifice. In order to determine if slightly super-heating the steam by adding heat to it would change the results, a coil of ¼ in. copper tubing about 15 in. long was inserted in the steam supply between valves *B* and *C*. The coil was heated with a blow torch sufficiently to raise the temperature of the steam one or two degrees. After a series of tests which gave the same result as without the heating operation, the tubing was removed and the tests continued. Thermometer *D* indicated approximately the saturation temperature in all tests. When a high vacuum was being carried at this point, a superheat of one to two degrees was observed.

Two makes of "radiator" valves were used during the tests but no difference in the results could be detected.

Results

The results of the tests on the orifices in the 3/32-in. thick plates with atmospheric pressure in the "radiator" are indicated by the curves in Figs. 4 and 5. The orifice plates were installed in the "radiator" end of the tailpiece. Fig. 4 shows the relation between flow and pressure drop through the orifice. It is well known that flow is not directly proportional to pressure drop. As shown by this figure, when the pressure drop through the orifice

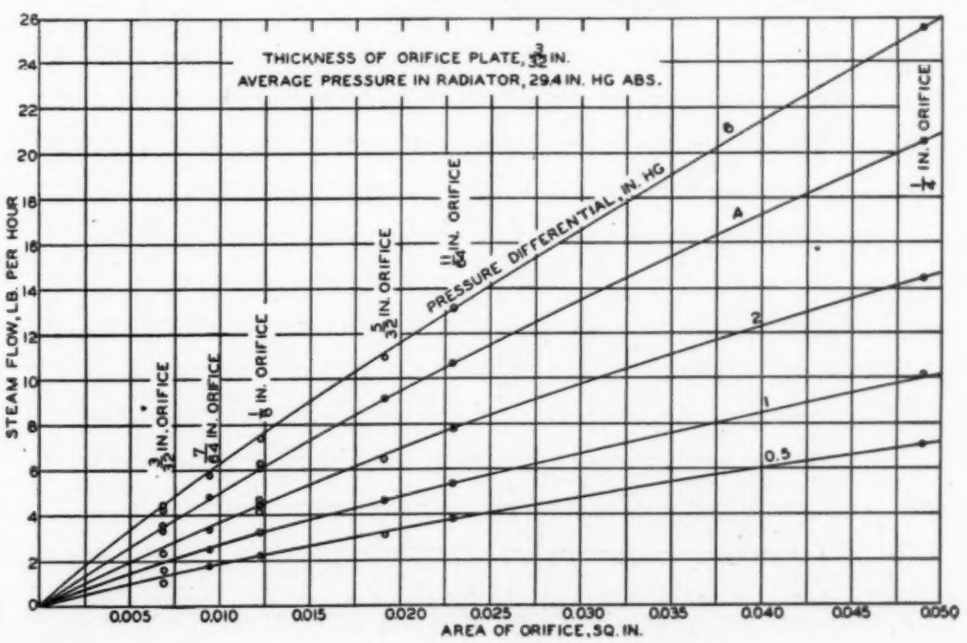
THICKNESS OF ORIFICE PLATE, 3/32 IN.
AVERAGE PRESSURE IN RADIATOR, 29.4 IN. HG ABS.
STEAM FLOW, LB. PER HOUR
PRESSURE DIFFERENTIAL, IN. HG
AREA OF ORIFICE, SQ. IN.

FIG. 5—FLOW OF STEAM VS. AREA OF ORIFICE FOR ORIFICES IN A PLATE 3/32 IN. THICK

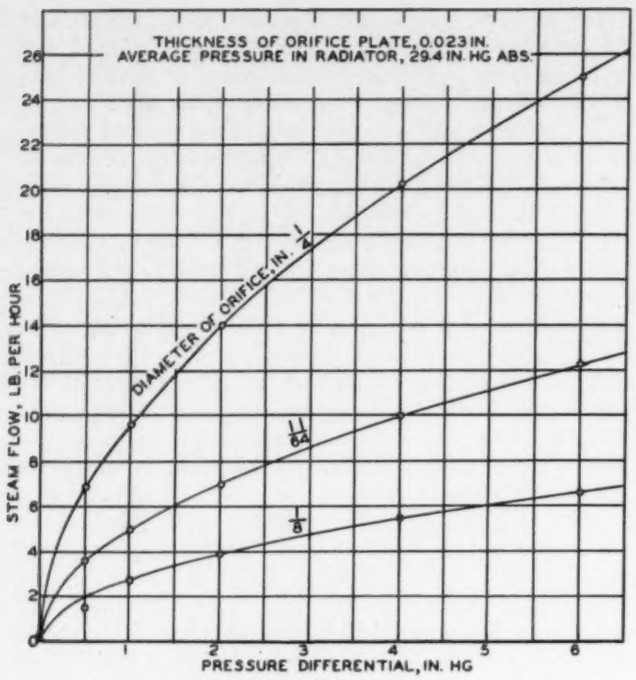

FIG. 6—FLOW OF STEAM VS. PRESSURE DROP THROUGH ORIFICE FOR ORIFICES IN A PLATE 0.023 IN. THICK

is low, small changes in pressure drop affect the flow to a greater extent than do the same changes in the region of high pressure drop.

Fig. 5 shows the relation between flow and cross-sectional area of orifice. It should be noted that the flow is not exactly proportional to the area. As the area of the orifice increases, the rate of increase of flow decreases.

The results of the tests on the orifices in the copper stampings 0.023-in. thick installed in the union end of the tail-piece with atmospheric pressure in the "radiator" are given by the curves in Figs. 6 and 7. The flow characteristics are similar to those for the thicker orifices except that the flow is slightly reduced.

Fig. 8 shows the relation between flow and orifice area for the orifices in the 3/32-in. thick plates when a vacuum of 20 in. of mercury existed in the "radiator." It should be noted that the flow was considerably reduced as compared with that at atmospheric pressure. The complete results for the two types of orifices with vacuums of 4, 8, 12 and 16 in. of mercury in the "radiator" are not included in this paper on account of lack of space.

The results obtained with a 6-in. pressure drop at these different conditions are given in Fig. 9. This figure shows the effect of initial pressure on the flow through an orifice. It indicates that for a given pressure drop through the orifice, the flow increases as the initial pressure is increased. The curves in Fig. 9 are for the orifices in the 3/32-in. thick plate with a pressure drop of 6 in. of mercury through them and are taken from the tests with pressures in the "radiator" varying from atmospheric to a vacuum of 20 in. of mercury.

The flow through a nozzle is given by the Moyer formula,

$$W = \frac{AP^{0.97}}{60} \qquad (1)$$

which is approximately the same as Grashof's formula,

$$W = 0.01654AP^{0.9700} \qquad (2)$$

where—

W is the weight of dry steam discharged in pounds per second,
A is the area of the orifice in square inches, and
P is the initial absolute pressure in pounds per square inch.

This formula holds only when the ratio of the final pressure to the initial pressure is less than 0.58. When the ratio of the final pressure to the initial pressure is greater than 0.58 the result given by the formula must be multiplied by a factor to obtain the actual flow. This factor is a function of the pressure ratio. The values

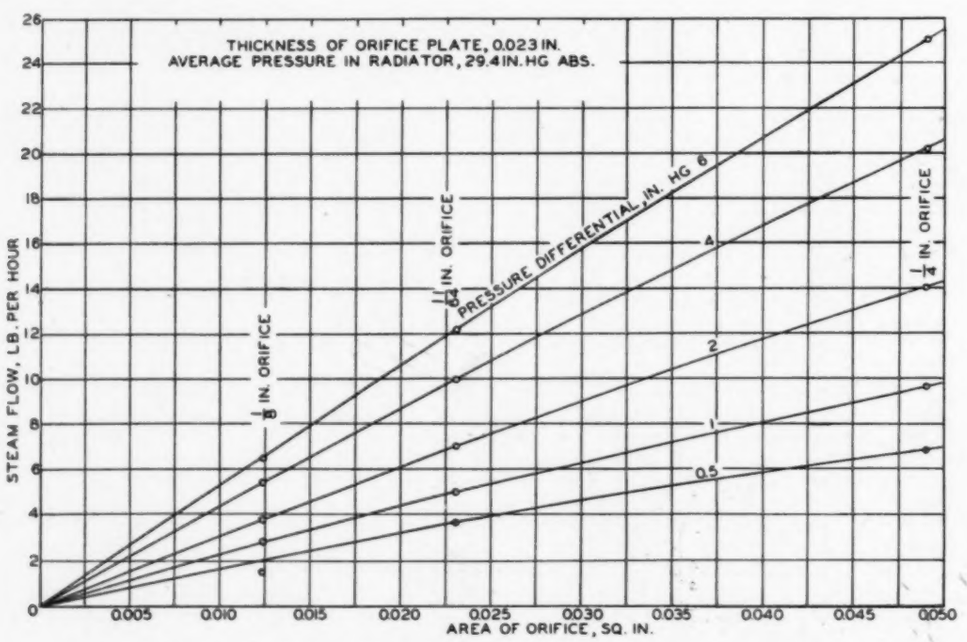

FIG. 7—FLOW OF STEAM VS. AREA OF ORIFICE FOR ORIFICES IN A PLATE 0.023 IN. THICK

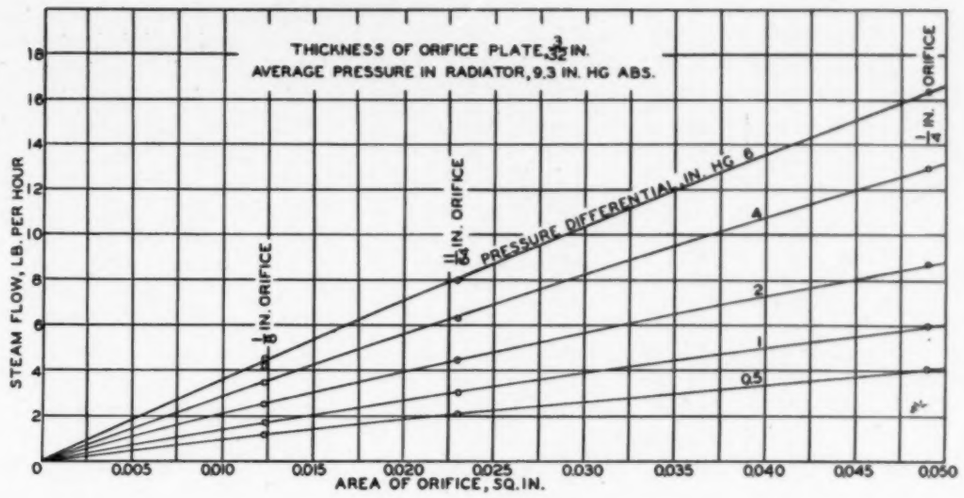

FIG. 8—FLOW OF STEAM AT SUB-ATMOSPHERIC PRESSURE VS. AREA OF ORIFICE

for it as determined by Rateau[3] are given by the upper curve in Fig. 10.

The flow through an orifice is less than through a nozzle and does not reach a maximum when the ratio of the final pressure to the initial pressure is 0.58. The flow continues to increase as the final pressure is reduced below 58 per cent of the initial pressure. The formula just discussed can also be used for calculating the flow through an orifice if the proper coefficients are used with it. The lower curve in Fig. 10 gives the coefficients for an orifice in a thin plate as determined by Rateau. The orifice which he tested had a diameter of 20.12 mm (0.792 in.) and was installed in a pipe having a diameter of 60 mm (2.4 in.). The discharge side of his orifice was cut away, making it a true thin-plate orifice.

[3] Experimental Researches on the Flow of Steam Through Nozzles and Orifices, A. Rateau, D. Van Nostrand Co., 1905.

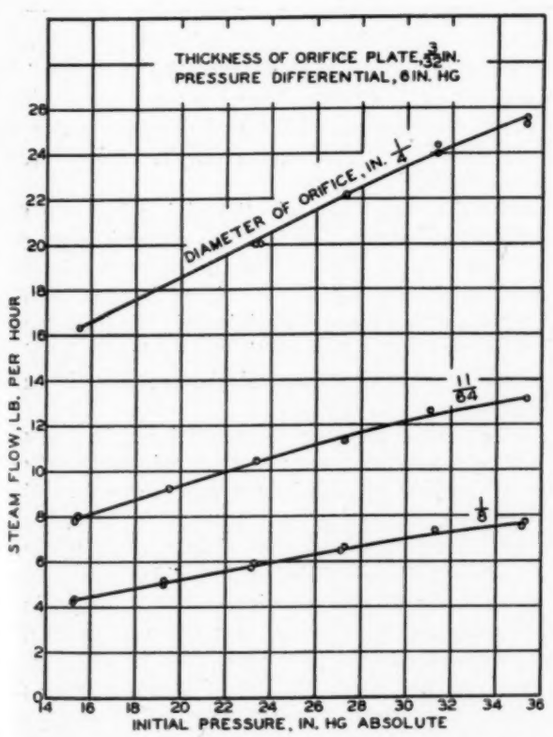
THICKNESS OF ORIFICE PLATE, 3/32 IN.
PRESSURE DIFFERENTIAL, 6 IN. HG
DIAMETER OF ORIFICE, IN. 1/4
11/64
1/8
STEAM FLOW, LB. PER HOUR
INITIAL PRESSURE, IN. HG ABSOLUTE

FIG. 9—EFFECT OF VARYING INITIAL PRESSURE WHILE KEEPING PRESSURE DROP THROUGH ORIFICE CONSTANT

The individual points on the curve sheet are from the tests reported in this paper. It will be noted that the points obtained from the tests on the thin copper orifices agree closely with the curve given by Rateau, whereas the flow through the thicker orifice is somewhat greater, approaching that through a nozzle as the ratio of orifice diameter to thickness of plate decreases. The point for the pressure ratio of 0.405 was determined from a special test.

During the various tests reported in Figs. 4 to 8, the barometric pressure varied between the limits of 29.15 and 29.54 in. of mercury. For most of the tests it averaged 29.4 in. The average during the series of tests shown by Fig. 8 was 29.3 in.

Table 1 shows the results of a series of tests made to determine the effect of moving the orifice from one end of the valve tail-piece to the other. In every case the flow of steam was greater with the orifice installed in the union end of the tail-piece. The data in columns 2 and 3 are from Figs. 4 to 7. All of the tests were run with atmospheric pressure in the "radiator." The average barometric pressure was 29.4 in. of mercury and the individual results were corrected to this pressure by using the curves in Fig. 9. During these tests the

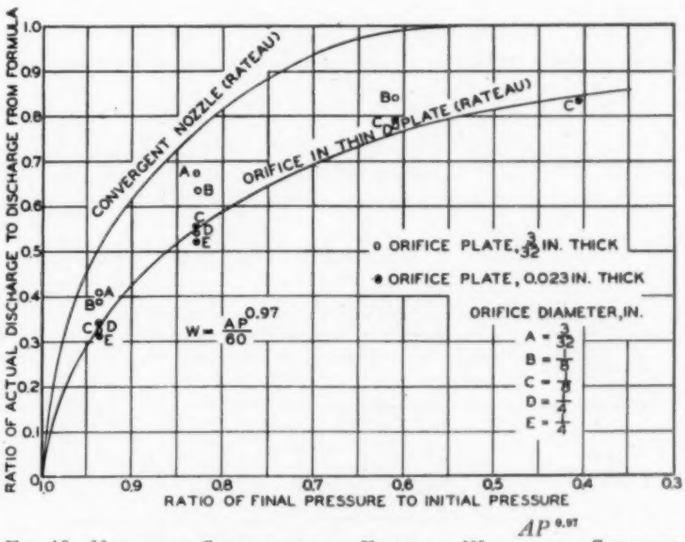
CONVERGENT NOZZLE (RATEAU)
ORIFICE IN THIN PLATE (RATEAU)
RATIO OF ACTUAL DISCHARGE TO DISCHARGE FROM FORMULA
RATIO OF FINAL PRESSURE TO INITIAL PRESSURE
o ORIFICE PLATE, 3/32 IN. THICK
• ORIFICE PLATE, 0.023 IN. THICK
ORIFICE DIAMETER, IN.
A = 3/32
B = 1/8
C = 11/64
D = 1/8
E = 1/4
W = AP^0.97 / 60

FIG. 10—VALUES OF COEFFICIENT FOR FORMULA, $W = \frac{AP^{0.97}}{60}$, SHOWING COMPARISON OF AUTHOR'S TESTS WITH CURVES BY RATEAU

TABLE 1—EFFECT OF POSITION OF ORIFICE ON STEAM FLOW[a]
Pounds per Hour Through ¼-in.

PRESSURE DROP THROUGH ORIFICE (IN. HG)	ORIFICE IN A PLATE 1/32 IN. THICK		ORIFICE IN A PLATE 0.023 IN. THICK	
	ORIFICE AT UNION	ORIFICE AT "RADIATOR" END	ORIFICE AT UNION	ORIFICE AT "RADIATOR" END
1	10.8	10.2	9.7	9.2
2	15.4	14.4	14.1	13.7
4	22.6	20.6	20.2	19 0
6	27.1	25.6	25.0	24.1

[a] "Radiator" pressure, 29.4 in. of mercury, absolute.

thin orifice plate was held in place by soldering it to the tail-piece.

The increased flow obtained with the orifice installed in the union end of the valve tail-piece can be explained by the probable formation of a region of reduced pressure in the tail-piece surrounding the vena contracta. This would occur if the flow filled the tail-piece at its discharge end. The action of the fluid steam would then partly evacuate the space formed. The pressure surrounding the vena contracta being less than the pressure in the radiator, the flow would correspond to this reduced discharge pressure rather than to the pressure in the "radiator."

Table 2 shows the results of a series of tests made to determine the effect of the thickness of the orifice on the flow of steam. All of these tests were run with the orifices in the "radiator" end of the tail-piece. The data in the first column are for the copper stamping soldered in the end of the tail-piece. The data in the second column are from the results given in Figs. 4 and 5. The data in the third column are from an orifice plate ¼ in. thick made for this test. These tests, which were run with atmospheric pressure in the "radiator" and corrected to 29.4 in. of mercury, absolute, show clearly that the flow increases with the thickness of the plate, indicating that the flow through a thick plate approaches that through a nozzle. Other investigators have found that for air, the maximum flow is obtained when the thickness of the plate equals the diameter of the orifice.[4]

Application

The tests demonstrate that it is possible accurately to control the flow of steam into a "radiator" by controlling the pressures on the two sides of an orifice located in the "radiator" valve. In a heating system, however, the pressure ahead of every "radiator" valve is not the same because of the pressure drop in the piping, and therefore the flow of steam through the orifices into the "radiators" nearest the source of supply will be greater than into the more distant "radiators." This inequality of flow can be minimized by using small orifices so that the pressure differential required across the orifice to produce a given flow will be large compared with the pressure drop through the piping.

It is not considered good practice to design a heating system for a total pressure drop through the piping of more than 8 oz. In a 40-story building in which the piping is designed for a pressure drop of 1 oz per 100 ft, the maximum difference in pressure between the nearest and farthest "radiators" might be of the order of 1 in. of mercury. This difference would occur only in extreme weather when the heating system is operating at maximum capacity. At other times, the pressure drop would of course be reduced because of the reduced flow of steam.

The allowable pressure drop through the orifice is limited by the noise, which must not be loud enough to annoy room occupants. It has been found by experience that a pressure drop through the orifice of 6 in. of mercury can be used without causing annoyance. If the orifices are made of such a size that a pressure differential of 6 in. of mercury is required to keep the "radiators" full of steam under normal conditions, then the sizes should be as given in Table 3. These sizes are based on atmospheric pressure in the "radiator" and orifice plates

TABLE 3—SIZES OF "RADIATOR" ORIFICES

ORIFICE DIAMETER (IN.)	SIZE OF "RADIATOR" (SQ FT)	ORIFICE DIAMETER (IN.)	SIZE OF "RADIATOR" (SQ FT)
1/32	18	5/32	61
1/16	24	11/64	70
3/32	30	7/32	80
1/8	37	13/64	91
5/32	44	1/4	102
11/64	52		

3/32 in. thick located in the "radiator" end of the valve tail-pieces. The condensation rate of the "radiator" is assumed to be ¼ lb of steam per hour per square foot of equivalent "radiation." If one of the other types of orifices is used instead, this table can be revised from the data given in this paper.

If the total pressure drop in the piping between the first and last "radiators" of a heating system is high, the required flow of steam into the more distant "radiators" can be obtained by using larger orifices in these "radiators." Another means of obtaining the desired steam flow into all of the "radiators" is to divide the heating system into zones and to control the pressure in each zone independently.

Table 4 gives the pressure differential required at the orifices to deliver the proper amount of steam to the "radiators" at different outdoor temperatures assuming that a "radiator" full of steam at atmospheric pressure will supply the required amount of heat when the outside temperature is zero and that the orifice sizes are as given in Table 3 and atmospheric pressure is maintained in the "radiators." This table was taken from Fig. 4 and is based on the assumption that if a 6 in. differential pressure gives the proper amount of steam at zero degrees, then the amount of steam required at any other temperature is proportional to the difference between 70 F and the outdoor temperature.

TABLE 2—EFFECT OF THICKNESS OF ORIFICE PLATE ON STEAM FLOW[a]
Pounds per Hour Through 11/64-in. Diameter Orifice

PRESSURE DROP THROUGH ORIFICE (IN. HG)	THICKNESS OF PLATE, 0.023 IN.	THICKNESS OF PLATE, 1/32 IN.	THICKNESS OF PLATE, ¼ IN.
1	4.8	5.4	6.4
2	7.0	7.8	8.6
4	9.7	10.7	11.9
6	12.0	13.2	14.1

[a] "Radiator" pressure, 29.4 in. of mercury, absolute.

[4] Steam and Gas Turbines, A. Stodola, Vol. I, pages 113-114. McGraw-Hill Book Co., Inc., 1927.

TABLE 4—REQUIRED PRESSURE DIFFERENTIAL ACROSS ORIFICES FOR DIFFERENT OUTDOOR TEMPERATURES[a]

Outdoor Temperature (Degrees Fahrenheit)	Pressure Differential (In. Hg)	Outdoor Temperature (Degrees Fahrenheit)	Pressure Differential (In. Hg.)
0	6	40	1.2
10	4.5	50	0.5
20	3.2	60	0.1
30	2.1		

[a] Atmospheric pressure in "radiator;" room temperature, 70 F.

The pressure in the heating system is usually controlled by adjusting a pressure reducing valve or a throttle valve in the steam main. This control valve must be set for a pressure sufficient to take care of the pressure drop in the piping between the control valve and the "radiators" in addition to the required pressure drop through the orifices. Table 4 shows that a very small differential pressure across the orifices must be used in mild weather. If the heating system is equipped with a vacuum pump, sub-atmospheric pressure can be carried in the "radiators." In that case a larger differential must be used to obtain the same steam flow (see Fig. 9). This is of advantage in that the flow can probably be more accurately regulated, as slight errors in adjusting the pressure will not affect the flow through the orifices as much as when smaller differentials are used.

The data presented in this paper have been based on the assumption that the system would be controlled by maintaining a constant pressure in the "radiators" and by varying the pressure in the steam mains. In vacuum systems, if desired, the steam pressure can be kept constant and the vacuum in the "radiators" varied to provide the required differential.

While the practice of dividing a building heating system into zones is desirable from the standpoint of control, it is not absolutely essential to the successful operation of an orifice system. One building having a gross volume of over 20,000,000 cu ft and containing about 270,000 sq ft of heating surface with orifices in the "radiators" is operating successfully with the heating system divided into three zones. In another case, a 46-story building with orifices in the "radiators" is operating successfully with one control valve for the entire building.

The "radiators" in an orifice heating system can be quickly filled with steam in the morning at the beginning of the warming-up period by using a sufficiently high differential pressure, so that the length of the warming-up period need not be appreciably longer than with an ordinary heating system.

A heating system equipped with "radiator" orifices has several advantages:

1. The building temperature can be controlled.
2. The amount of steam used for heating is reduced.
3. The building temperature control is in the hands of the building management, where it can be properly supervised.
4. Excessive opening of windows and the resulting loss of heat is avoided.
5. The cost of making and installing orifices is very low. (Those shown in Fig. 1 at *B* and *C* are easy to install and the stampings are inexpensive to make in quantities.)

It is desirable to use manometers instead of pressure gages in adjusting the pressure at the control valve as accurate adjustment of pressures is essential to the successful operation of an orifice system. Orifice systems should be installed only in those buildings where there is a member of the operating staff available who will give close attention to adjustment of pressures. An electrical thermometer system giving temperatures throughout the building is of assistance in operation.

Conclusions

The flow of steam through thin-plate orifices installed in the unions of "radiator" valves is substantially the same as reported by Rateau for a thin-plate orifice in a straight run of pipe. The flow can be calculated from the formula $W = \frac{AP^{0.97}}{60}$ by multiplying the result by a factor obtained from Rateau's curve for thin-plate orifices. For thicker orifices the results indicate a higher rate of flow than given by this curve.

"Radiator" orifices can be inexpensively made and installed. When an orifice system is properly operated, it is possible to obtain highly satisfactory control of building temperature, resulting in economy in the use of steam for heating. Table 5 shows the savings made in

TABLE 5—RESULTS OBTAINED BY INSTALLING ORIFICES IN "RADIATORS"

	Building *A*		Building *B*	
	Before Installing Orifices	After Installing Orifices	Before Installing Orifices	After Installing Orifices
Period covered	1-31-29 to 5-31-29	1-31-30 to 5-31-30	11-30-29 to 3-31-30	11-29-30 to 3-31-31
Average temperature, degrees Fahrenheit	42.7	44.1	29.4	31.1
Degree-days (65 degree Fahrenheit base)	2736	2572	4308	4136
Steam for heating, pounds	39,040,400	32,958,860	21,818,300	17,558,800
Steam for heating, pounds per degree-day	14,300	12,810	5,065	4,240
Steam saved by installing orifices, per cent		10.4		16.3

	Type	"Radiation" (Sq Ft)	Net Heated Space Inside Building (Cu Ft)
Building *A*	Office Building	267,560	16,437,510
Building *B*	Office Building	93,190	6,858,360

two large Detroit office buildings by installing orifices in the "radiator" valves. Both of these buildings were very carefully operated before the installation of orifices and the steam consumption at that time was about the minimum that could be obtained without using some type of control. The savings reported, therefore, are savings due alone to the installation of orifices. The steam consumption reported is for heating only and does not include steam for water heating and other purposes. About 11 per cent of the space in building *B* is occupied by a bank which is air-conditioned, so that the steam consumption of that part of the building was not affected by the installation of orifices.

Acknowledgment

The tests reported in this paper were made in connection with the work of The Detroit Edison Co. in instructing its steam heating customers in the economical utilization of steam for heating. The authors are indebted to C. W. Signor and E. T. Smith of the Central Heating Department, who conducted the tests and who contributed many helpful suggestions.

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Fluid Metering in Practice, by E. S. Smith, Jr. (Mechanical Enginering, Nov. 1930, pp. 968-970).

Bibliography of the Bureau of Mines Coal Investigations

The numerous contributions to the literature of coal made by the members of the technical staff of the United States Bureau of Mines, Department of Commerce are listed in Technical Paper 493, Bibliography of United States Bureau of Mines Investigations on Coal and Its Products, 1910-1930, by A. C. Fieldner and M. W. von Bernewitz, recently published. One of the principal functions of the Bureau of Mines is investigation of the efficient production and use of solid, liquid and gaseous fuels and their by-products and publication of the findings thereon. The study of fuels, particularly coal, was begun in 1903 by the technologic branch of the United States Geological Survey, which was merged into the Bureau of Mines when it was organized in 1910. This bibliography on coal and its products includes publications issued by the Bureau of Mines and those written by its staff for the technical press, which includes the chemical and engineering periodicals of scientific bodies. Under this head are included reports of work done jointly with States and colleges.

The papers on coal and its products are separated into the class of publications and arranged alphabetically by authors. The publications of the Bureau of Mines comprise bulletins, technical papers, miners' circulars, reports of investigations, information circulars, and special reports.

In addition, this bibliography includes references to papers on the economics and statistics of coal and coke issued by the economic branch of the bureau. This branch was organized shortly after July 1, 1925, when the former mineral resources division of the United States Geological Survey was transferred by Executive order to the Bureau of Mines. The economics branch of this bureau thus inherited certain records of the Geological Survey and assumed the responsibility of carrying forward the annual volumes, Mineral Resources of the United States, which from 1880 to 1923 had been issued by the survey. Beginning with 1924, therefore, the Mineral Resources reports have appeared under the auspices of the Bureau of Mines. As the Mineral Resources volumes from 1917 to 1923 contained many important data on coal economics, the Director of the Geological Survey has permitted the inclusion in this bibliography of references to these volumes and to other publications of the staff of the coal statistics section during their service with the survey. The economics branch also issues a series of current reports on coal and coke.

Copies of Technical Paper 493 may be obtained from the Superintendent of Documents, Government Printing Office, Washington, D. C.

Predetermining the Airation of Industrial Buildings

By W. C. Randall[1] and E. W. Conover[2], Detroit, Mich.
MEMBERS

THE method presented in this paper for predetermining airation (the flow of air into, through and out of buildings due to the natural forces, the wind and temperature difference) is the result of efforts to shorten the somewhat lengthy but more technically correct method developed in the research investigations conducted under the direction of Prof. J. E. Emswiler of the Department of Engineering Research of the University of Michigan.

The correct method is to combine the heads created by these two forces each window opening, or group of openings, occurring in the different zones of pressure, suction, and normal atmospheric pressure. These various zones are determined by the length, width and height of the structure and the direction of the wind, and may also be influenced by the distance from and size of neighboring buildings. Such factors must and still will be used when greater accuracy than can be secured by the simplified method is essential or desired. That a shorter method could be used was clearly indicated after solving several hundred problems. The method suggested herein was found to be reasonably accurate, at least within the percentage of error of other assumptions that have to be made because of lack of knowledge, or because of efforts to simplify the problem.

It is hoped that the method presented in this paper will make it possible to estimate airation in advance of construction with sufficient accuracy for practical purposse. It should be possible thereby to design the fenestration for any desired movement of air, and to make comparisons of the probable results from more than one design.

It is not to be presumed that this can be done accurately for every situation that arises. However, it can be done for certain simpler cases, and the performance under more complicated circumstances inferred with better judgment.

The forces of nature are always present, whether it is desirous that they be put to work or not. A better knowledge of the natural flow of air may explain some of the idiosyncrasies of mechanical systems, the success or failure of which depend to some extent upon whether or not the natural forces are co-operating with or opposing them, and as a result, more dependable installations can be designed.

In outline, this simplified method consists of three steps: (1) a determination of the air flow due to temperature difference, (2) the flow due to the wind, and (3) a combination of the flow created by these two forces acting together.

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For presentation at the Semi-annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Swampscott, Mass., June, 1931.

Flow Due to Temperature Difference

The air within a building absorbs much of the heat used in or generated by manufacturing processes, and must be carried off and be replaced by cooler outside air if conditions are to be maintained which will allow the workmen to continue their labors without undue discomfort. This is usually accomplished by providing upper openings through which the heated air may escape and lower openings through which the cooler air may enter.

The factors which influence the volume of air moved by temperature difference alone are:

1. The amount of heat given off.
2. The vertical distance between inlet and outlet.
3. The areas of inlet and outlet openings.

These three factors were adequately discussed in a previous paper[3] based upon this research work and a method was given therein for determining the air flow through a building induced by this force alone.

The difference in temperature existing between the inside and outside air is determined by the amount of heat given off inside the building, and the amount of air flowing through it. The first requirement, then, is to estimate the amount of heat to be carried off by the air in, for example, Btu per minute. This can be determined, in the case of forge shops and the like, from the amount of fuel burned which, in turn, is based upon the production capacity for which the building is being designed. In the case of foundries, the heat given off by the metal in cooling from the molten state can be used. In some cases, not all of the heat may be dissipated to the air, such as that carried off through stacks, that absorbed by the sand in foundries and that taken out by removing castings from the building while still hot. But a fair estimate of the amount to be removed by the air can usually be made.

The next step is to select the temperature difference to be maintained. The selected inside temperature should be low enough for the comfort of the workmen. In the summer time, when outside temperatures are high, the inside-outside temperature difference must not be great. Consequently, the greatest amount of air must be passed through the building at that time, and the airation characteristics determined for that condition.

Knowing the amount of heat to be removed and having selected a desirable temperature difference, the amount of air to be passed through the building per minute to maintain comfortable conditions can be determined by means of the following equation:

$$H = 0.0175QD \qquad (1)$$

[3] The Neutral Zone in Ventilation, by J. E. Emswiler (A. S. H. V. E. TRANSACTIONS, Vol. 32, 1926).

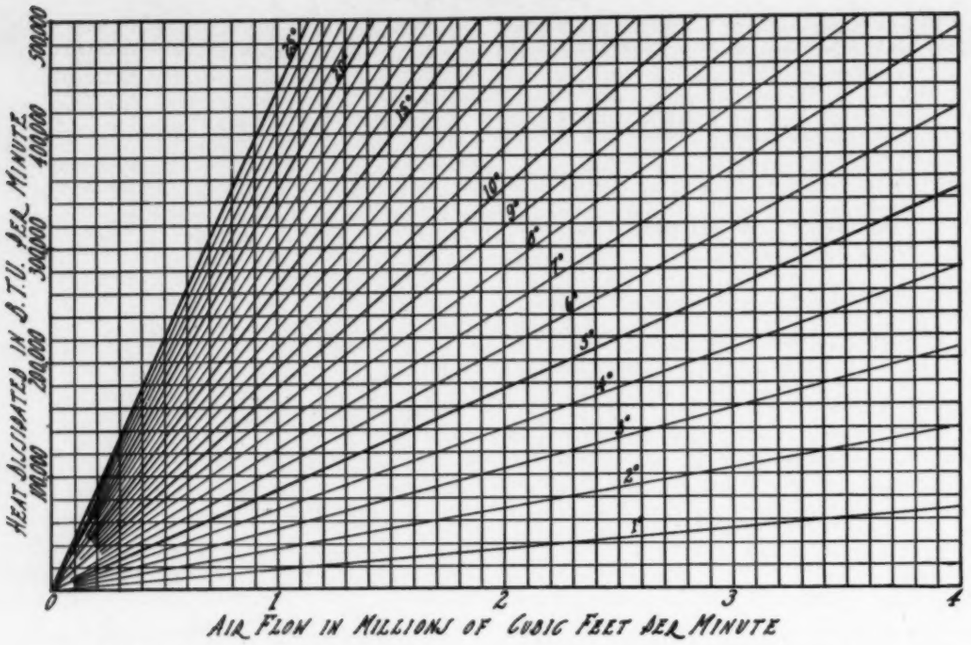

FIG. 1—AIR FLOW REQUIRED TO MAINTAIN DESIRED TEMPERATURE DIFFERENCE

where

H = heat in Btu per minute
Q = air flow in cubic feet per minute
D = inside-outside temperature difference, degrees Fahrenheit

Equation (1) is solved graphically in Fig. 1, thus making it possible to determine readily the air flow necessary to maintain any desired temperature difference for a given heat emission.

For a given temperature difference (D) and vertical distance between openings, the outlets will exhaust a definite amount of air. The elemental building dimensions, such as width, length and height, are usually determined by the nature and the amount of the product and the methods of handling production. Therefore, the approximate location of the windows is fairly well established, and the height of the outlets above the inlets is known.

It was stated in a previous paper,[4] that the inlets and outlets should be about equal for the greatest flow per unit of combined areas and, since this is the ideal construction, this condition will be considered first. Knowing the temperature difference to be maintained and the vertical distance between inlets and outlets, the flow per square foot of inlet or outlet can be determined by the following equation:

$$Q = 60\sqrt{\frac{MD}{2RT}} \qquad (2)$$

where

Q = flow in cubic feet per minute per square foot of inlet or outlet (when of equal area)
M = vertical distance between inlets and outlets in feet
D = inside-outside temperature difference, degrees Fahrenheit
T = average absolute temperature (460 F plus average between inside and outside, say 550 F for summer)

R = resistance of opening $= \dfrac{1}{2gA^2C^2}$

g = acceleration due to gravity = approximately 32.2 ft. per second per second
A = opening area in square feet (in this case 1)
C = coefficient of resistance of the opening (0.65)

Substituting the foregoing values of R and T in equation (2),

$$Q = 9.4\sqrt{MD} \qquad (2a)$$

Fig. 2 has been plotted from equation (2a), and from this figure the flow per square foot of inlet or outlet opening, when of equal areas, may be read directly for any given distance between the openings and any temperature difference. If for a certain building (1) the amount of air to be passed through, (2) the temperature difference to be maintained, and (3) the distance between inlets and outlets, are known, the amount of window opening to be installed as inlets (with a corresponding amount for outlets) may be ascertained by

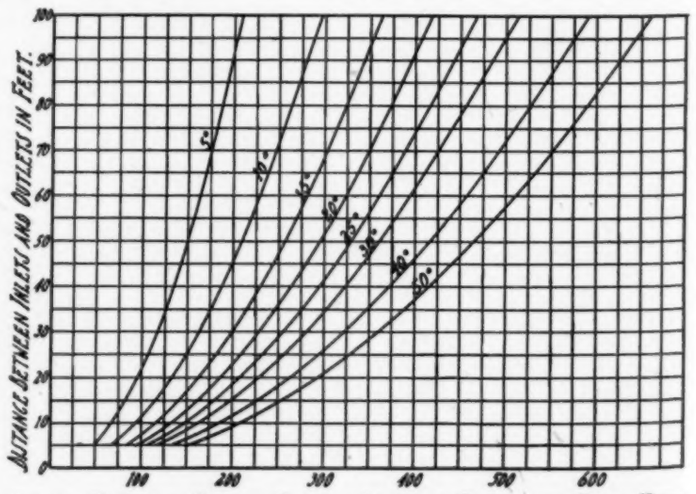

FIG. 2—FLOW PER SQUARE FOOT OF INLET OR OUTLET IN CUBIC FEET PER MINUTE—INLET AND OUTLET AREAS EQUAL

[4] Airation of Industrial Buildings, by W. C. Randall (A. S. H. V. E. TRANSACTIONS, Vol. 34, 1928).

dividing the flow per square foot of opening, as determined from Fig. 2, into the total amount of air necessary to be moved.

Although the greatest flow per unit area of opening is obtained when inlets and outlets are equal, it is frequently found that a building has greater possible outlet openings than inlets, or vice versa. Equal openings will result in a certain amount of air flow. Increasing the area of one over the other will increase this flow, but not in proportion to the added area. The increase in flow produced by excess of one opening over the other is indicated by Fig. 3. For example, if the outlet area is double that of the inlet, the flow will be about 26½ per cent greater than would be the case if it equaled the inlet. This curve has been obtained by solving problems according to the method outlined in the paper The Neutral Zone in Ventilation previously mentioned, thus giving a substitute for determining the neutral zone when openings are unequally divided.

When designing a building, it might be found that the amount of inlet opening calculated to be necessary could not be incorporated in the plans without major changes. Sometimes it is possible to lower the level of the inlets or raise that of the outlets, or both, which would give an increased head and a consequent increased flow per square foot of opening. If this cannot be done, then the alternative is to install as many lower sidewall openings as possible for inflow, and to increase the upper outlet openings to give the required flow, or vice versa.

Fig. 4 shows the combination of inlet and outlet areas that will give the same results as those secured by equal openings. For example, assume that 1,000 sq ft of inlet openings and an equal amount of outlets are necessary to give the desired results in a certain instance. On laying out the plans, it is found that but 800 sq ft of opening can be accommodated in the lower sidewall. This being 80 per cent of that required, it is found from Fig. 4 that the upper openings will need to be 190 per cent of 800 or 1,520 sq ft. This gives a total of 2,320 sq ft required for the equivalent performance of 2,000 sq ft equally divided between inlet and outlet.

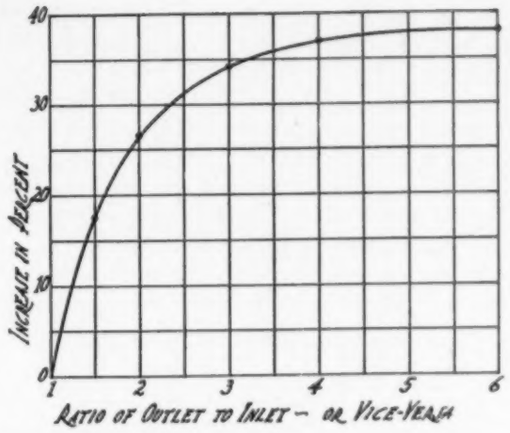

Fig. 3—Increase in Flow Caused by Excess of One Opening Over the Other

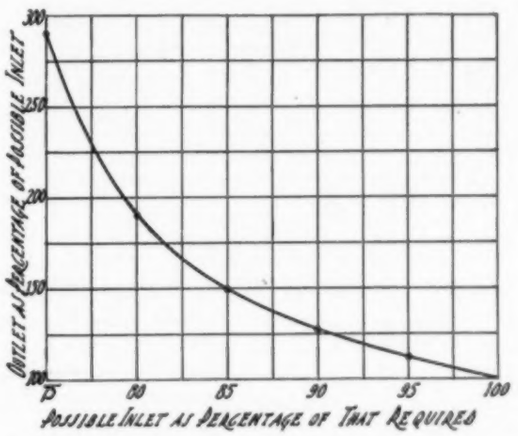

Fig. 4—Combinations of Inlet and Outlet Openings to Produce the Same Results as for Equal Inlet and Outlet Openings

Flow Due to the Wind

In machine shops, assembly buildings and the like, where little heat is generated, the airation is produced chiefly by the wind. The requirements for such buildings are somewhat less than for foundries, forge shops, heat-treatment buildings, and similar types of buildings where smoke, gases, etc., prevail. When the wind is producing the airation, many factors enter into the correct solution of the problem. For simplification, the buildings would first be divided into two classes, namely, those with a monitor, or monitors, and those without, since there is an increased flow caused by monitors which usually place openings in the suction zones created by the wind hopping over the building.

The main factors which influence the air flow are the size and shape of the building, the velocity and relative direction of the wind and the size and location of window openings. It is believed that a simple and workable method must be based upon the average results produced by these factors.

To determine the window openings required in a given building, it is necessary to select first the desired air changes or air movement. It is also necessary to know the velocity and direction of the prevailing winds for the particular location being considered. These usually can be obtained from the nearest local office of the U. S. Weather Bureau.

With this information available, the amount of window opening to be provided can be determined by simple percentages. If the building has no monitor and the wind will blow perpendicular to the side, the area of the openings to be installed in the windward side may be found by dividing the desired air flow in cubic feet per minute by 50 per cent of the velocity of the wind in feet per minute. If blowing diagonally, 30 per cent should be used, but this will give the combined area in one side and one end. Naturally, corresponding amounts will be necessary in the opposite side and end to act as outlets.

If a monitor is contemplated, this will put additional openings in the suction zone caused by the wind jumping over the building and, consequently, the air flow will be increased. If the monitor openings correspond in area to those in the sidewall below, then the desired flow should be divided by 60 and 35 per cent of the velocity of the wind, respectively, for wind directions perpendicular and diagonal to the side. If a greater area is pro-

vided in the monitor than in the sidewall, these percentages will be increased and, if less, they will be decreased.

The foregoing percentages usually will take care of all cases when considering side or diagonal winds against all buildings not obstructed by others. If the prevailing wind will blow against the end of the contemplated building and the structure is other than a square one, the percentages given by Fig. 5 will apply.

When end winds are blowing, it usually is found that some sidewall openings, other than those in the direct

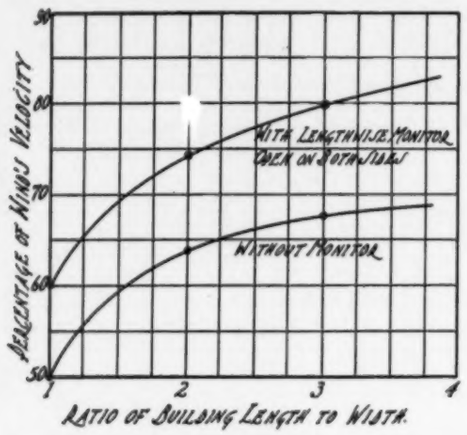

FIG. 5—PERCENTAGES FOR WIND AGAINST END

path of the wind, will provide inflow. This is due to a greater demand for air at openings in the areas of suction than can be supplied by the windward openings, thus pulling air in at openings beyond the direct influence of the wind. This will occur at monitor openings as well as sidewall, but in arriving at the prediction figures shown, a deduction has been made for what is considered ineffective inflow at monitor openings on the assumption that some of it is short circuited and of no material benefit in the working zone. Thus, in predetermining the air flow by this method, the results represent more closely the effective flow rather than the total air passing through the building.

The curves of Fig. 5 are based upon the ratio of the length and width of the building, and will give the area of the windows to install in the end elevation. A corresponding fenestration in the bays of the sidewalls and monitor, if any, will complete the design.

Flow Due to Combined Wind and Temperature Difference

To determine the size of window openings that will provide a predetermined flow of air through a building in which considerable heat is produced, it is desirable to use the method described for flow due to temperature difference alone, in order that adequate openings will be provided to take care of the days when little or no wind is blowing.

In some instances, this will show an amount that is impossible or impractical to install. Account must then be taken of the action of the wind and the design based on the combined forces. It is recognized, as previously stated, that the correct method is to combine the forces or heads caused by each, rather than to add the flows. However, it was found that there was some relation between the ratio of the calculated flows produced by each agent separately and the flow computed by adding the heads.

Fig. 6 shows that the flow created by the combined forces is a multiple of that due to temperature difference alone. This graph represents the results obtained chiefly from solving problems involving three building heights, five wind velocities, four temperature differences, three building lengths and all wind directions. It has been found that this method will give results that agree, in most cases, with those obtained by combining the forces, the maximum variation being about 15 per cent.

To apply this method, it is necessary first to compute the flow that would be produced by each force if acting alone. The percentage each is of the sum is then determined. Using the temperature-flow percentage as an index, the flow due to the combined forces which is a multiple of that calculated for temperature difference alone, may be obtained from Fig. 6. For example, assume that the computed flow due to temperature difference alone is 30 per cent of the added amounts and that due to the wind, 70 per cent. The flow due to the combined forces, as found from Fig. 6, would be 2.45 times that computed for temperature difference alone.

With the greater flow of air caused by the two forces acting together, it is evident that more heat will be carried out of the building and, consequently, the inside temperature will drop below that experienced when no wind is blowing. The magnitude of this drop can be determined from Fig. 1. For example, assume that the heat to be dissipated by the air is 200,000 Btu per minute, and that the windows are designed to maintain a temperature difference of 15 deg. The air flow induced by temperature difference alone is 750,000 cfm. Assume also that the effect of the wind increases this to 1,000,000 cfm. The amount of heat dissipated is not changed

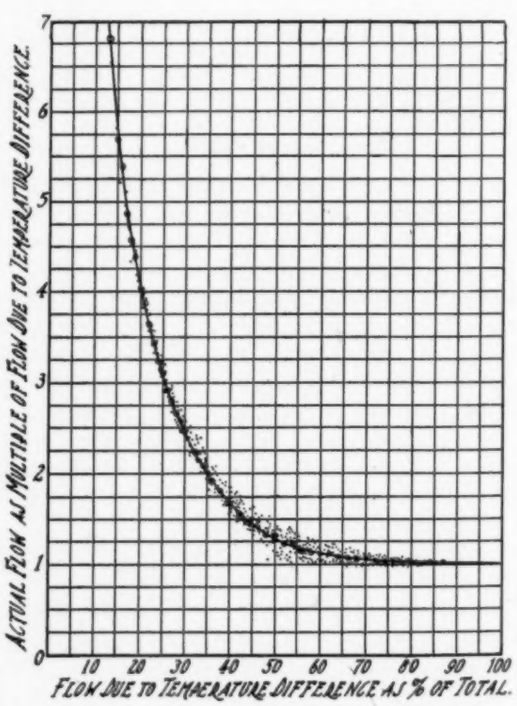

FIG. 6—DETERMINATION OF FLOW CAUSED BY COMBINED FORCES OF WIND AND TEMPERATURE DIFFERENCE

so the intersection of the "200,000 Btu" line with the "1,000,000 cfm" line gives the resultant inside temperature as about 11½ deg above that outside instead of the 15 deg when no wind is blowing.

Solution of a Typical Problem

A typical problem will illustrate this procedure. Let it be assumed that the building outline has been determined to be as shown in Fig. 7, and that the prevailing conditions are as follows:

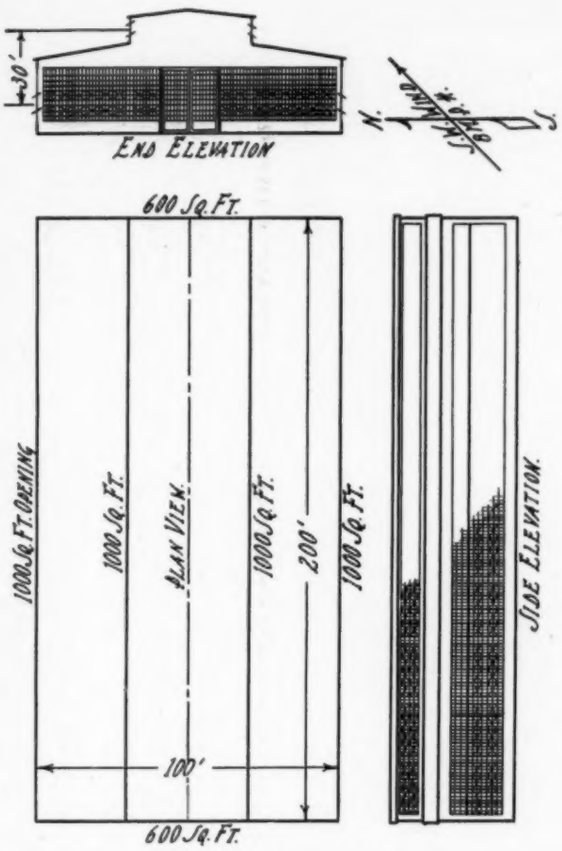

FIG. 7—TYPICAL INDUSTRIAL BUILDING

Drop forge shop
Building dimensions: 100 ft × 200 ft × 30 ft
Cubical contents: 600,000 cu ft
Height outlet above inlet: 30 ft
Fuel: 30 gal fuel oil per hour
Desired temperature difference: 10 deg
Wind diagonal at 8 mph

Computations for Temperature Difference Only

The heat to be removed $= \frac{30 \text{ gal} \times 7.75 \text{ lb} \times 18{,}000 \text{ Btu}}{60} =$ 69,750 Btu per minute.

This amount of heat will require about 400,000 cfm of air (Fig. 1) or 40 air changes per hour to maintain the desired temperature difference. With 30 ft between inlet and outlet and a 10-deg temperature difference, the flow per square foot of inlet or outlet is 165 cfm (Fig. 2), and $\frac{400{,}000}{165} = 2{,}400$ sq ft of opening required in the sidewall and an equal amount in the monitor. Say that only 2,000 sq ft can be readily accommodated in the two sides of the monitor. This is 83 per cent of 2,400. The inlets must then be 160 per cent of 2,000 or 3,200 sq ft (Fig. 4). With the building twice as long as it is wide, this quantity would be divided into about 1,000 sq ft for each side and 600 sq ft in each end.

Computations for Wind Only

Wind velocity = 8 mph (diagonal) = 704 fpm
Window openings in path of wind = 1,600 sq ft (one side and one end)
$0.35 \times 704 \times 1{,}600 = 394{,}240$ cfm

Computations for Combined Forces

When the wind is blowing, the windward side of the monitor should be closed, which gives the conditions of 1,000 sq ft in the upper openings, and 3,200 sq ft in the lower, a ratio of 3.2 to 1. The flow due to temperature difference must then be determined for that condition and the computation would be—

$1{,}000 \times 165 \times 1.35 = 222{,}750$ cfm (Fig. 3)

Adding the two flows:

Temperature difference	= 222,750 =	36 per cent
Wind	= 394,240 =	64 per cent
Total	616,990 =	100 per cent

The computed flow would then be about:
1.9 x 222,750 = 423,225 cfm (Fig. 6)

The original flow due to temperature difference was 400,000 cfm with all windows open. The effect of the wind is to increase the flow to 423,225 cfm which, according to Fig. 1, reduces the temperature difference from the original 10 deg to about 9 deg, even though one half of the monitor windows is closed.

When little or no wind is blowing, all windows may be opened wide to utilize the full effect of temperature difference. However, when there is a side or diagonal wind, the tendency is for air to enter on the windward side of the monitor and thus to decrease the amount that the leeward side can exhaust from below. By using judgment in the manipulation of the windows and closing upper inflow openings, the windows are utilized to the best advantage for effectively airating the building.

Some times the problem is one of effectively removing smoke, fumes or dust rather than of removing heat. This introduces an indeterminate factor impossible of mathematical solution unless experience has demonstrated just what air movement is necessary in the particular case in question. Even when considering the removal of heat, the airation is based on the judgment of what will be a satisfactory temperature difference. Predetermination methods, therefore, are predicated upon the selection of an air movement or temperature difference rather than upon definitely stated requirements.

It is occasionally found that the computed opening areas are small compared to those normally placed in a building. While they may provide the computed amount of air under the conditions for which the problem is solved, it is advisable to protect the design against possible change in use at a later date, and also to secure sufficient air for extremely hot days by providing wide openings through which the workmen also may see.

Checks with Field Tests

The degree of accuracy of the predetermination methods given herein is indicated in the comparison between the predictions and some of the field tests that were made. These comparisons are shown in Table 1. In this table the computed flows are determined for the conditions of wind velocity and direction, temperature difference and window openings as observed.

It is evident that the observed flow was in excess of that computed in all of these cases. The explanation for this is that the predetermination methods are based upon the effective flow through the building, while the observed flows usually include some so-called short circuiting, that is, air flowing in at certain high openings and out again at others without passing through the working zone.

Table 1—Comparison Between Observed and Computed Values

Building	Test No.	Wind Velocity	Temperature Difference (F)	Flow in Cu Ft per Min		Disagreement in Per Cent
				Observed	Computed	
U. of M. Foundry Ann Arbor, Mich.	382	8.5	25	20,000	16,810	16
U. of M. Foundry	385	7.8	6	21,400	15,010	30
Riter-Conley Co. Leetsdale, Pa.	261	3.2	0	335,000	293,715	12
D.S.P. Co. Spring Plant Detroit, Mich.	8	15.2	5	2,225,500	2,126,500	5
New Departure Co. Bristol, Conn.	2	4.3	18	1,670,000	1,602,000	4

Summary and Conclusions

This paper is intended as a companion paper to The Neutral Zone in Ventilation, by J. E. Emswiler, and Airation of Industrial Buildings, by W. C. Randall, in which the general principles of airation were discussed at length. Therefore, the authors have confined themselves to the presentation of a method for computing either the amount of window opening to install in a building to obtain a predetermined airation, or the airation to be expected from a given fenestration.

The size and shape of an industrial building are usually adapted to the desired production capacity and the nature of the industry. The fenestration should also be determined by these same items, if the natural forces are to be depended upon to produce the airation. The data in this paper make it possible to determine the window openings necessary to produce a selected air movement, or a desired temperature difference when the general building dimensions and heat emission, as well as the velocity and relative direction of the prevailing winds are known. It also makes it possible to compare several designs for airation results and thus to select the most desirable one.

Components of Coal

Coals are composed essentially of two visibly different major components which are present in different types of coal in greatly varying proportions, according to Dr. Reinhardt Thiessen, research chemist on the constitution of coal at the Pittsburgh Experiment Station, of the U. S. Bureau of Mines, Department of Commerce. These two components are designated as *anthraxylon* and *attritus*. Anthraxylon is a structurally simple component, while attritus is a complex component, composed of a large number of various plant entities. In addition to these two major components, there is a third, minor component called fusain or *mineral charcoal*, which is found in most coals in relatively small proportions, although occasionally layers are found containing considerable amounts. Its origin and formation is a much debated question upon which there is no agreement. Apparently fusain is derived from woody tissue of which the structure is often well preserved. It is higher in carbon content and lower in volatile matter than the other components, and has no coking properties.

Anthraxylon is the undisintegrated woody tissue of plants. It shows under the microscope a more or less visible homogeneous cell structure. Bright coal bands are essentially anthraxylon, being the stems, branches, twigs and roots of plants that were buried in the vegetable debris during the peat stage coalified and subsequently flattened, but yet remaining as unit constituents in the final coal bed. The anthraxylon strips or lenses vary in thickness from a fraction of a millimeter to several centimeters. In general, they have a smooth, black to a highly lustrous appearance according to the rank of the coal—the higher the rank the higher the gloss. The coking properties of coal are due mainly to the anthraxylon components. However, all anthraxylon will not coke. It possesses this property only in coals within certain ranges of rank or degrees of coalification.

Anthraxylon and attritus may be associated in coals or in particular bands of coal in all proportions from practically 100 per cent anthraxylon to 100 per cent attritus. Cannel and boghead coals consist almost entirely of attritus, whereas the ordinary *bright* coals are composed of varying proportions of anthraxylon and translucent attritus, and *splint* coals are composed of lesser amounts of anthraxylon associated with opaque attritus.

PITTSBURGH EXPERIMENT STATION OF THE U. S. BUREAU OF MINES WHERE THE RESEARCH LABORATORY OF THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS IS LOCATED

A New Era in A. S. H.V. E. Research Begins

HERE is news—good news—for A. S. H. V. E. members as well as the entire heating and ventilating profession and industry. A new and comprehensive policy developed from the experience of 10 years of cooperative research becomes effective immediately.

Centralized control is continued through the Chairman of the Committee on Research with the assistance of a Technical Advisor.

The Director of the Research Laboratory will have charge of all investigations at the Society's Laboratory in the Bureau of Mines Experiment Station, Pittsburgh.

All matters of policy will be determined by the Committee on Research as a whole and all business affairs remain in the hands of the Chairman and the Executive Committee. A Technical Advisory Committee will be assigned to each research problem and will report to the chairman of the Committee on Research.

C. V. Haynes, Chairman of the Committee on Research, announces that in line with the policy approved by the Committee, Arthur C. Willard, Professor of heating and ventilation and head of the Department of Mechanical Engineering, University of Illinois, has been retained as Technical Advisor for Research.

Professor Willard's work in heating and ventilating research

PROF. A. C. WILLARD
Technical Advisor for Research

C. V. HAYNES, Chairman
Committee on Research

F. C. HOUGHTEN, Director
Research Laboratory

F. B. Rowley
U. of Minnesota

G. L. Larson
U. of Wisconsin

A. P. Kratz
U. of Illinois

C. P. Yaglou
Harvard

L. C. Seeley
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Supervising A. S. H. V. E. Research at Cooperating Institutions

makes him outstanding in the profession and his advice and suggestions will be invaluable in extending the scope of Society research activities and in coordinating the data for practical application by the heating and ventilating profession.

Prof. Arthur C. Willard, past president of the Society, is outstanding as an engineer, educator, and research investigator and to him belongs the credit for many discoveries which have added to the comfort, health and efficiency of people throughout the country.

In the field of research Professor Willard is recognized as an outstanding authority and perhaps his best known work has been the warm air furnace work covered in the past 10 years for the *National Warm Air Heating Association* and the investigations conducted for the *National Boiler and Radiator Manufacturers Association* and the *Illinois Master Plumbers Association*.

Some of the experimental work that he has supervised includes heat transmission through building materials, heat emission from steam radiators, flow of steam in heating mains and risers, studies of indirect radiators, industrial unit heaters, effect of radiator enclosures and covers and studies of hot water heating.

Professor Willard is the author of numerous bulletins and circulars published by the University of Illinois and is co-author with L. A. Harding of two books on Mechanical Equipment of Buildings. He has been a prolific writer for the technical press and has contributed numerous scientific papers to engineering societies.

One of the outstanding accomplishments in his career of public service was his work as consultant on ventilation to the late Clifford M. Holland, Chief Engineer for the New York and New Jersey Bridge and Tunnel Commissions, responsible for the building of the Holland Tunnel for vehicles between New York and New Jersey. Professor Willard supervised a great deal of experimental work in connection with the ventilation methods used, which was conducted at the University of Illinois and at Pittsburgh, Pa. Since 1921 Professor Willard has been a consultant for the U. S. Bureau of Mines and in 1925 he became a consultant to the U. S. War Department, Chemical Warfare Service Section, and more recently has acted as consultant for the Architect of the Capitol at Washington in planning the ventilation of the House and Senate Chambers, and for the Chicago Subway.

Professor Willard was born in Washington D. C., and after receiving his preparatory education in the public schools of the District of Columbia, he attended Massachusetts Institute of Technology and was graduated in 1904.

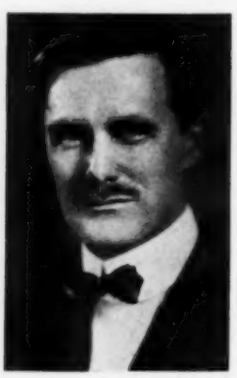
Upon graduation he commenced his teaching career in California and in 1906 came to George Washington University as assistant professor of mechanical engineering.

From 1909 to 1913 he served the government as sanitary and heating engineer at the War Department in the office of the Chief of Q. M. C.

In 1913 he joined the staff of the University of Illinois as assistant professor of heating and ventilation and his progress during the years since that time has placed him in charge of the department and brought him many honors in engineering circles.

During the war Professor Willard acted as consultant on heat-

L. E. Davies
Armour Institute
of Technology



S. E. Dibble
Carnegie Institute
of Technology

A. S. Langsdorf
Washington University

F. E. Giesecke
Texas A. & M.

Supervising A. S. H. V. E. Research at Cooperating Institutions

ing, cooking, and laundry equipment for the U. S. Army and then returned to his duties in the Mechanical Engineering Laboratory at the University of Illinois. Since 1920 he has been head of the Department of Mechanical Engineering.

In the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS he has served on many of the technical committees, including the Guide Publication Committee, Committee on Standards of Ventilation, Committee on Code of Minimum Requirements for the Heating and Ventilation of Buildings and has also held the offices of President, First Vice-President and Second Vice-President. In addition to this, he has served on many Council Committees and has been chairman of the Executive Committee, Publication Committee, Meeting Program Committee, and for a number of years has served as a member of the Committee on Research.

The Committee on Research feels that with the growing activities, it is fortunate to have the assistance of Prof. Willard as Technical Advisor for Research.

A renewal of the contract with the U. S. Bureau of Mines has been approved and the investigations in Pittsburgh will be directly in charge of F. C. Houghten, Director of the Research Laboratory.

The latest cooperative agreement has been made with the University of Illinois and the problems assigned to that institution are under the immediate supervision of Prof. A. P. Kratz.

Cooperative agreements are now in effect with the following institutions:

1. United States Bureau of Mines.
2. Armour Institute of Technology.
3. Carnegie Institute of Technology.
4. Harvard University.
5. Texas Agric. and Mech. College.
6. University of Illinois.
7. University of Minnesota.
8. University of Wisconsin.
9. Washington University, St. Louis.
10. Yale University.

Twelve problems are being studied at present either at the Research Laboratory in Pittsburgh or at the nine cooperating universities, and other investigations will be undertaken as approved by the Committee on Research as finances permit:

1. Atmospheric dust and air cleaning devices.
2. Air conditions and their relation to living comfort.
3. Physiological effects of atmospheric environment on human beings at work.
4. Heat transmission through building materials.
5. Conductivity studies on concrete.
6. Heat capacity of concrete slabs.
7. Heating effect of the sun on building roofs and walls.
8. Infiltration of air through walls and building openings.
9. Boiler testing with oil fuel and rating of oil burners.
10. Measurement of air flow through registers and grilles.
11. Flow characteristics of copper tubing for steam and hot-water heating.
12. Performance characteristics of various enclosed types of cast iron and non-ferrous radiators under actual heating service conditions.

The Society retains the distinction of being the only professional engineering organization to maintain and operate its own Research Laboratory and after completing 10 years work at Pittsburgh in cooperation with the U. S. Bureau of Mines, it should be gratifying to Society members to know that they are directly contributing 40 per cent of their annual dues to scientific work that has received world-wide recognition because of its scientific and practical value both in and outside of engineering circles.

With but little outside assistance, the heating and ventilating profession and industry has financed the Society's research in order to obtain the needed fundamental facts to assure progress in the art. The number of dollars spent at our own Research Laboratory in Pittsburgh and in the cooperating institutions during the entire existence of the Laboratory, exceeding $300,000, represents only a portion of the total amount expended, as the government furnishes without cost to the Society such important items as space, heat, light, technical assistance and shop service. This also applies to the cooperating institutions who supply funds in equal amount to those allotted by the Society.

Expenditures for A. S. H. V. E. Research

1919	$13,358.39	1925	$17,623.82
1920	20,225.12	1926	26,607.49
1921	27,664.60	1927	†35,324.83
1922	*11,863.33	1928	32,017.90
1923	33,235.43	1929	33,683.15
1924	21,519.91	1930	40,486.26

* Fiscal year changed from Aug. 1 to July 30, to calendar year Jan. 1-Dec. 30.
† Cooperative agreements started.

Encouraged by the successful operation of research investigations during the 10-year period in cooperation with the Bureau of Mines, in Pittsburgh, and for the past three years with several other institutions it was the belief of the Committee on Research this year, that the aim for 1931 would be to spend the funds most effectively and estimated expenditures have been budgeted accordingly.

Budget Expense 1931

Expense for Committee on Research and work at A. S. H. V. E. Laboratory, Pittsburgh, and in Cooperating Institutions $43,750

No reduction in research activity is contemplated, but the program for 1931 will be to get the most per dollar spent in the way of scientific data and practical results.

A survey undertaken by the Chairman of the Committee on Research and the Executive Committee, immediately after the 37th Annual Meeting in Pittsburgh, disclosed the urgent need of carrying on the present program without curtailment in 1931 and revealed the interesting fact that scientific research will receive the active financial support that it merits in good times and bad.

Contributions from the industry for A. S. H. V. E. Research should total $25,000 to help cover the cost of the 1931 Program.

Those who have made pledges for 1931 are:

Manufacturers

American Air Filter Co.
American Blower Corp.
Barnes and Jones
E. K. Campbell
Carrier Corp.
Celotex Co.
Hoffman Specialty Co.
Ilg Electric Ventilating Co.
Masonite Corp.
Nash Engineering Co.
Powers Regulator Co.
H. B. Smith Co.
Taco Heater
Trane Co.
United Engineers and Constructors.

Associations

American Oil Burner Association
Associated Copper Tubing Manufacturers
Heating and Piping Contractors National Assn.
Chicago Ventilating Contractors Em. Assn.
National Lumber Manufacturers Association

SKIPPER JONES
Finance

JOHN WEBB
Reception and Registration

TOM MCCOY
Transportation

JOE BRINTON
Entertainment

ALFRED KELLOGG
Hotel and Meeting

COL. BOYDEN
General Chairman

LESLIE CLOUGH
Publicity

ED DUSOSSOIT
Sports

FRANK TUTTLE
Golf

MRS. DUSOSSOIT
Ladies' Entertainment

All Set for Swampscott

June 22-25, 1931

"Old Ironsides"

SUMMER days are here and A. S. H. V. E. members who gather at Swampscott, Mass., June 22 to 25, will find a wonderful program of technical subjects provided for discussion and a sports and recreation program arranged for their pleasure which will keep every minute occupied.

Swampscott is a delightful place where business and pleasure can be combined and this thought has been uppermost in the minds of the Committee on Arrangements when the program was made up.

The New Ocean House, one of the show places on the North Shore, is the meeting headquarters. It faces the ocean and from the broad deck, as Skipper Jones would call the veranda, the guests see the ocean across a beautiful bathing beach.

Colonel Boyden has recruited his army of assistants sometime ago and has been drilling them in their duties which is to provide for the comfort of A. S. H. V. E. members from the moment they arrive in Swampscott. The first general order issued for the meeting was "Everybody is to have a good time at Swampscott."

With this idea in mind, the Colonel selected his assistants with care and assigned Tom McCoy to keep the transportation under control, secured Frank Tuttle to administer the golf tournaments because of his knowledge of traps, and Jim Brinton to provide a breezy banquet entertainment. The colonel retained a consulting engineer, Alfred Kellogg, to plan the meetings and survey hotel facilities and contracted with Ed Dusossoit to lay out a program of sports and with Mrs. E. A. to keep the ladies entertained and contented.

John Webb will be on deck to receive the guests and give them a hearty greeting. Barnacle Bill Jones (not the radio sailor) will keep the finances on the course in Chairman Boyden's opinion and Leslie Clough was assigned to do publicity. His picture galleries have already produced some unconventional poses of his fellow committee men.

Here are shown the General Arrangements Committee Chairmen when they were not occupied with their daily duties and it will be noted that the majority are addicted to golf.

All of them will be on hand at Swampscott to meet the visiting members of the Society and each will wear a distinguishing red, white and blue badge and are all set to see that everyone has a good time.

In making arrangements to go to Swampscott it should be remembered that transportation may be by train, plane, motor or steamship (from New York).

Fare and one-half from all points in the United States and Canada will be in effect and the rail costs from Chapter cities is indicated in the following table:

	One-Way Fare Swampscott	Fare and One-half Convention Swampscott
Buffalo, N. Y.	$ 18.29	$ 27.44
Chicago, Ill.	37.00	55.50
Cleveland, Ohio	24.83	37.25
Denver, Colo.	74.38	111.57
Detroit, Mich.	27.29	40.94
Kansas City, Mo.	53.64	80.46
Milwaukee, Wis.	40.06	60.09
Minneapolis, Minn.	51.66	77.49
New York City	8.73	13.10
Philadelphia, Pa.	12.87	19.31
Pittsburgh, Pa.	24.55	36.83
Seattle, Wash.	111.41	*158.61
St. Louis, Mo.	43.93	65.90
Toronto, Ont.	21.86	32.79
Los Angeles, Calif.	114.04	*158.61
San Francisco, Calif.	114.04	*158.61

*Summer Tourist rate.

Identification certificates with which fare and one-half tickets may be purchased at the member's home station will be mailed from the Society headquarters by the Secretary in ample time for the meeting. Tickets for the entire family may be purchased with one certificate. Sample form is shown.

Rail service from New York is via New York, New Haven and Hartford with trains leaving at frequent intervals from Grand Central Station. New Haven trains arrive in Boston at *South Station* and members should taxi to *North Station* for Boston and Maine trains which will take them to Swampscott unless they are traveling with a special group from one of the Chapters.

Members coming from the west via Albany, N. Y., by routing themselves over the Boston and Maine R. R. will arrive in *North Station* and may make immediate connections with B. & M. trains to Swampscott. Those who route themselves via New York and sail from Pier 19 on the palatial steamers *New York* and *Boston* through Cape Cod Canal will arrive in Boston at India Wharf and should taxi to *North Station* for Swampscott trains.

Those who desire to make the summer meeting week an attractive vacation trip and decide to come to Swampscott by

FANEUIL HALL AND CUSTOM HOUSE, BOSTON

FOUNDERS' MEMORIAL, BOSTON COMMON

OLD SOUTH CHURCH, BOSTON

PAUL REVERE'S HOUSE, BOSTON

OLD STATE HOUSE, BOSTON

BOSTON TEA PARTY TABLET

MINUTE MAN STATUE, CONCORD

MINUTE MAN STATUE AND CONCORD BRIDGE

SAILORS' MEMORIAL, GLOUCESTER

Historic Places to be Visited by A. S. H. V. E. Members and Guests on Motor Trips

"House of the Seven Gables," Salem

automobile may do so via New York and then over Route 1 (the Shore Line), through New Haven, New London and Providence *or* by way of Bear Mountain Bridge on Route 6, through Danbury, Waterbury, Hartford, Springfield and then Route 20 through Worcester.

Motorists coming from the north or west through Albany, N. Y., will find two fine routes, No. 2 (the Mohawk Trail), through North Adams, Greenfield and Fitchburg *or* Route 20 (Jacobs Ladder), through Pittsfield, Springfield and Worcester. Many places of historic interest lie along these routes and the city of Boston provides an unusually attractive place for the sightseer. Roads in Massachusetts are excellent and many side trips are available from Swampscott in case motorists care to explore.

Registration will begin on Monday, June 22, in the South Lobby of the New Ocean House and the remainder of the day will be devoted to various Committee Meetings.

At noon a luncheon for the Officers and authors of papers will be held and during the afternoon the Council and several Committees will meet. All events will be held on daylight saving time.

Three technical sessions are to be held from 9:30 a.m. until 12:30 p.m. with the remainder of each day, Tuesday, Wednesday and Thursday available for sports, inspection trips or sightseeing.

The detailed program of technical sessions as shown indicates that the papers are of high quality on subjects of vital interest to heating and ventilating engineers. Much of the work to be reported upon results from the research program, undertaken by the Society both at Pittsburgh and in the cooperating Universities.

Several committees' reports and codes will provide interesting discussions.

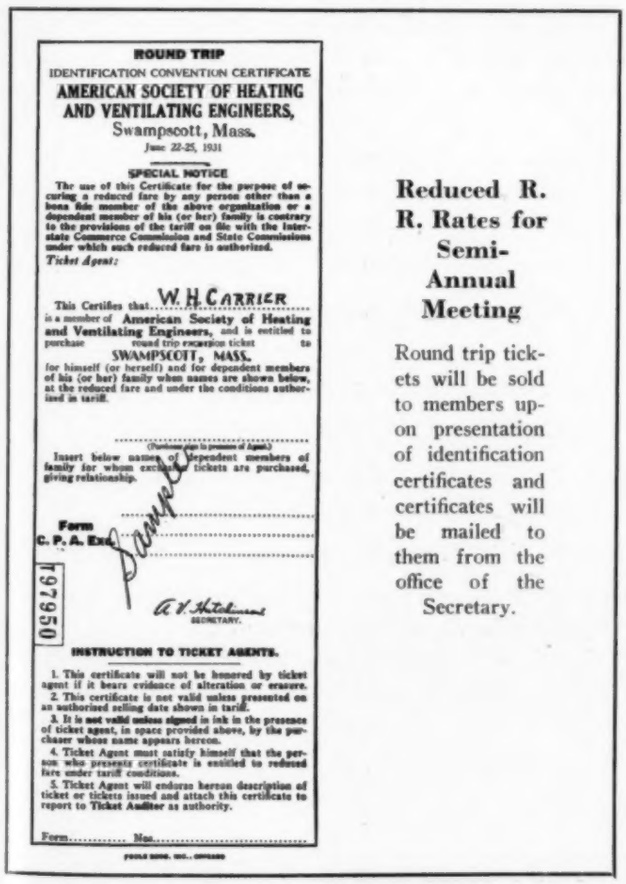
The festivities commence with an informal reception and dance on Monday night, June 22, in the ballroom of the New Ocean House and everyone is urged to attend.

The special events for the ladies commence on Tuesday morning at 10 o'clock when a clock golf tournament will be held on the lawn of the New Ocean House. At 1:30 p.m. a combination coach and boat trip will be taken along Massachusetts famed North Shore, as far north as Gloucester. While the ladies and those who do not play golf are seeing the sights of the North Shore, the golfing members of the Society will participate in a Kickers' Handicap Tournament at the United Shoe Machinery Country Club in Beverly. Motor cars will leave the hotel promptly at 2 o'clock. At 8 o'clock Tuesday evening a card party will be held in the New Ocean House for ladies and those who desire may dance in the ballroom.

A special feature for the ladies will be a trip to Boston at 9 o'clock Wednesday morning where they will visit *Old Ironsides* and ancient land marks.

At 2 o'clock in the afternoon autos will leave the hotel for the Winchester Country Club, where the members will play an 18-hole tournament for the Research Cup.

On Wednesday evening at 7 o'clock the Semi-Annual Banquet and Dance will be held at the New Ocean House.



ROUND TRIP

IDENTIFICATION CONVENTION CERTIFICATE

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS,
Swampscott, Mass.
June 22-25, 1931

SPECIAL NOTICE

The use of this Certificate for the purpose of securing a reduced fare by any person other than a bona fide member of the above organization or a dependent member of his (or her) family is contrary to the provisions of the tariff on file with the Interstate Commerce Commission and State Commissions under which such reduced fare is authorized.

Ticket Agent:

This Certifies that W. H. CARRIER
is a member of American Society of Heating and Ventilating Engineers, and is entitled to purchase round trip excursion ticket to
SWAMPSCOTT, MASS.
for himself (or herself) and for dependent members of his (or her) family whose names are shown below, at the reduced fare and under the conditions authorized in tariff.

(Purchaser sign in presence of Agent.)

Insert below names of dependent members of family for whom excursion tickets are purchased, giving relationship.

Form C. P. A. Ex.

197950

A. V. Stutsman
SECRETARY.

INSTRUCTION TO TICKET AGENTS.

1. This certificate will not be honored by ticket agent if it bears evidence of alteration or erasure.
2. This certificate is not valid unless presented on an authorized selling date shown in tariff.
3. It is not valid unless signed in ink in the presence of ticket agent, in space provided above, by the purchaser whose name appears hereon.
4. Ticket Agent must satisfy himself that the person who presents certificate is entitled to reduced fare under tariff conditions.
5. Ticket Agent will endorse hereon description of ticket or tickets issued and attach this certificate to report to Ticket Auditor as authority.

Form..... Mar.....

Reduced R. R. Rates for Semi-Annual Meeting

Round trip tickets will be sold to members upon presentation of identification certificates and certificates will be mailed to them from the office of the Secretary.

At 9 o'clock Thursday morning a special ladies' golf tournament will be held on the hotel golf course and at 2 o'clock in the afternoon all members and ladies are expected to be ready for a motor trip through Lexington, Concord and Salem.

In view of the many transportation problems confronting the local committee, it is requested that each member who attends, register in the manner requested at the earliest possible time, in order that adequate transportation facilities may be available.

The cooperation of the members attending the meeting will be greatly appreciated by the local committee as it is the earnest wish of every Massachusetts member that the visiting members and guests have a good time at Swampscott, June 22 to 25, 1931.

Gloucester Fishing Fleet

Method of Choosing Location of, Financing and Conducting Meetings of the Society

Resolved: That inasmuch as the Annual and Semi-Annual Meetings of the Society come under the jurisdiction of the Council, the following rules governing the handling of such meetings be adopted by the Council and published in the Journal of the Society at least twice during every year, preferably just prior to each meeting:

1—The Council will select the city in which the Annual or Semi-Annual Meeting is to be held, giving due consideration to the invitations received from Chapters or members as well as to the advisability of so distributing those meetings as to make them of the greatest advantage to the general membership, and to reduce as far as possible the expense of members attending.

2—That an appropriation be made to cover the entertainment or local expenses, incurred in connection with the meeting not exceeding $500.00, the regular meeting expense to be taken care of by the General Fund of the Society in the regular way.

3—That no registration fee or compulsory obligations of any nature be imposed on members or guests.

4—That the purchase of tickets for banquets or for any other form of entertainment that may be provided be entirely voluntary.

5—That the grouping of features and the same of tickets for group features be discouraged.

6—That the raising of Funds from manufacturers of heating apparatus be discouraged.

7—That the display of samples, or of literature, advertising the product of any manufacturer in any way, shape or form, be not permitted at the booths, registration desk, or in or about the meetings.

8—That the distribution of trade papers be entirely at the discretion of the committee in charge.

9—That the local Chapter, or local members, be empowered to form a General Committee with such subcommittees as may be required to handle the details of transportation, hotel accommodations, entertainment, finance, etc., and that this General Committee be requested to confer frequently with the Council, through the Secretary of the Society, and to make frequent reports on progress in connection with the various matters being handled by them.

10—That the arrangements of elaborate and costly entertainment features be discouraged.

Adopted at Council Meeting, January 29, 1926.

Program
Semi-Annual Meeting

American Society of Heating and Ventilating Engineers

New Ocean House, Swampscott, Mass., June 22-25, 1931

TECHNICAL

Monday, June 22, 1931

9:30 a. m. Registration.
12:30 p. m. Luncheon (Officers, Council and Authors).
2:00 p. m. Council Meeting.
3:00 p. m. Meeting Committee on Research.
3:00 p. m. Meeting Nominating Committee.
4:00 p. m. Meeting Guide Publication Committee.

Tuesday, June 23, 1931

8:30 a. m. Registration.
9:30 a. m. Technical Session.
Greeting by D. S. Boyden, Chairman Committee on Arrangements.
Response by Pres. W. H. Carrier.
Report of Committee on Increase of Membership, by C. W. Farrar, Chairman.
Flow of Steam through Orifices into Radiators, by S. S. Sanford and C. B. Sprenger.
Friction Heads in One-Inch Standard Cast-Iron Tees, by F. E. Giesecke and W. H. Badgett.
Sizing Pipes and Orifices for Gravity Hot Water Heating Systems, by E. G. Smith.
Using Gas-Vapor Mixtures for Heating Purposes, by C. A. Dunham.
Report of Guide Publication Committee, by D. S. Boyden, Chairman.
Report of Committee on Testing and Rating Unit Ventilators, by John Howatt, Chairman.
7:00 p. m. Conference of Chapter Representatives.

Wednesday, June 24, 1931

9:30 a. m. Technical Session.
Essential Elements for Determining Heating Plant Requirements, by F. B. Rowley.
Report of Committee on Rating and Testing Concealed Gravity Type Radiation, by R. N. Trane, Chairman.
Performance Characteristics of Cast-Iron and Non-Ferrous Radiators, by A. C. Willard.
Experiments to Determine Heating Effect Factor, by R. N. Trane and C. J. Scanlan.
Study of Performance Characteristics of Oil Burners and Low Pressure Heating Boilers, by L. E. Seeley and E. J. Tavanlar.

Thursday, June 25, 1931

9:30 a. m. Technical Session.
Report of Committee on Ventilation Standards, by W. H. Driscoll, Chairman.
Report of Committee on Research, by C. V. Haynes, Chairman.
Reports of Technical Advisory Committees.
Heat and Moisture Losses from Men at Work and Application to Air Conditioning Problems, by F. C. Houghten, W. W. Teague, W. E. Miller and W. P. Yant.
Infiltration through Double Hung Wood Windows, by G. L. Larson, D. W. Nelson and R. W. Kubasta.
The Measurement of the Flow of Air through Registers and Grilles, by L. E. Davies.
Predetermining the Airation of Industrial Buildings, by W. C. Randall and E. W. Conover.

ENTERTAINMENT

Monday, June 22, 1931

12:30 p. m. Luncheon (Officers, Council and Authors).
8:30 p. m. Reception and Dance.

Tuesday, June 23, 1931

10:00 a. m. Clock Golf.
1:30 p.m. Coach and Boat Trip to Marblehead and Gloucester including the famous North Shore Drive.
2:00 p. m. Golf—Kickers' Handicap.
8:00 p. m. Bridge Party for Ladies.

Wednesday, June 24, 1931

9:00 a. m. Motor Trip to Historic Boston and Boston Navy Yard where an inspection of the *Frigate Constitution* or *Old Ironsides* is planned.
2:00 p. m. Golf—Research Cup Tournament.
7:00 p. m. Banquet and Dance at New Ocean House.

Thursday, June 25, 1931

9:00 a. m. Special Golf Tournament—Ladies.
2:00 p. m. Trip to Historic Lexington, Concord and Salem.

9th Hole Winchester Country Club

Local Chapter Reports

Cleveland

March 27, 1931. Following the routine business Pres. F. H. Morris introduced H. F. Marshall, assistant sales manager of Warren Webster & Co., who gave an illustrated lecture on the Developments in Steam Heating and Central Control Systems.

In his opening remarks Mr. Marshall presented a lantern slide showing the wide variations in outdoor temperature which take place from day to day during the period of the heating season and the percentage of heat required during such times. He also pointed out the importance of central control of the heat supply to radiators in order to prevent overheating of the rooms and to secure economy of operation.

At the conclusion of Mr. Marshall's interesting talk the members and guests of the Cleveland Chapter showed their appreciation by extending a rising vote of thanks.

Michigan

April 20, 1931.—Ninety-two members and guests of the Michigan Chapter of the Society met at the Cadillac Athletic Club for the regular monthly meeting, at which Pres. E. E. Dubry presided.

Following some announcements concerning the slate of officers for the year 1931-32, President Dubry introduced H. P. Dolan, supervising engineer of the Detroit Board of Education, who invited the members of the Michigan Chapter to attend the next Maintenance Employees' Meeting, at the Northern High School, May 7.

The guest speaker of the evening was Prof. A. C. Willard, head of the department of mechanical engineering, University of Illinois, whose subject was Recent Research in Heating and Ventilation at the University of Illinois. The speaker covered completely the work which his department has done since 1918 with reference to the development of warm air furnaces. Every phase of engineering which would possibly be applied to warm air heating and ventilation was touched upon in Professor Willard's talk.

The next speaker, E. B. Langenberg, Vice-president of Langenberg Mfg. Co., St. Louis, and past-president of the *Warm Air Heating Association,* was then introduced by President Dubry. Mr. Langenberg based his talk on the necessity of the warm air heating industry taking advantage of and rigidly following the results which have been obtained in the testing laboratory of the University of Illinois. He stressed the necessity of coordination among the different factors in the industry and expressed the thought that wonderful possibilities existed in the industry in the near future.

A discussion of great interest followed, and the Chapter officers expressed the opinion that no meeting in recent months has been more interesting or instructive than this one. The entire membership of the Chapter extended their appreciation to Professor Willard and Mr. Langenberg.

Minnesota

April 10, 1931.—The Chapter was honored at this meeting by the presence of W. H. Carrier and A. V. Hutchinson, President and Secretary of the A. S. H. V. E. The meeting was held at the Men's Union Bldg., at the University of Minnesota, and was attended by 120 members and guests.

A brief business session followed the regular dinner, after which Prof. F. B. Rowley introduced Mr. Carrier, who spoke on Servicing the Human Power Plant. The scope of President Carrier's talk covered the physiological reactions of the body to temperature, humidity and air motion. He brought out the fact that air conditioning is successfully used for many types of industrial work and expressed the opinion that in years to come this will be in general use for office buildings, homes, etc.

After a rising vote of thanks was given to President Carrier the meeting was brought to a close.

New York

April 20, 1931. The principal speaker at this last business meeting of the Chapter was W. H. Carrier, President of the Society. Mr. Carrier's absorbing talk on Servicing the Human Power Plant which was illustrated by slides, was greatly appreciatetd by the attentive audience. His discussion was based, to a large extent, on investigations at the Research Laboratory of the Society, and at the Harvard School of Public Health.

In considering the requirements of the human body, Mr. Carrier stated that the most useful method of approach for the engineer is to compare it with a power plant, which, in fact, it is. It is the function of the human body to take fuel into the alimentary canal, which may be considered the pulverizer and magazine for the fuel, and to burn this fuel after being distributed through the arteries to the tissues of the body, which are the furnace. The air for combustion, and the products of combustion, are both handled by a combination of forced and induced draft through the lungs and respiratory system. The energy produced by the combustion of the fuel with the oxygen of the air is transferred into mechanical energy and heat with a considerable measure of efficiency. This transformation, as well as the attained efficiency, is in strict accordance with the laws of thermodynamics which indicates that only a portion of the available heat energy can be transformed into work, while the greater portion must always be expelled from the system in the form of heat at a lower potential; that is, the entropy of the system always increases.

Just as in a power plant, it is necessary not only to supply oxygen to the body and to remove the products of combustion, carbon dioxide and water vapor, but it also is necessary to remove the large proportion of unavailable heat which is present in every thermodynamic power cycle.

In the steam power plant, to which the human body may be compared, there must be a condenser in which the heat is removed by water or other heat conveying medium. In the case of the body, this heat conveyor is the blood. This cooling liquid must be pumped through the condenser or heat absorber. In the body the heart is the pump. When the heat in the power plant has been transferred from the condenser to the water, means must be provided for again cooling this water before recirculation. In the power plant operating on a closed cycle this water is cooled by radiation, conduction, and evaporation in a cooling tower through which the cooling water must be circulated in the closed cycle. Similarly in the human body, there must be a cooling tower—the skin (and to a much smaller extent, the lungs). It is a function of heating and ventilating engineers to provide for the removal of the waste products of the human power plant, the waste heat and the products of combustion.

In the early days of the art, the principal function of ventilation was considered to be the removal of the waste products of combustion, and little thought was given to ventilation as a means of removing the waste heat.

According to present available information there are never insufficient quantities of oxygen present or an excess of waste products in the air of a ventilated room when the requirements of air for removing the heat from the body are met; that is, the requirements for operating the cooling tower of a human power plant are far in excess of those for supplying the oxygen

chemically required for combustion just as it is in the mechanical power plant.

In ventilation or air conditioning for human comfort the prime consideration is, therefore, the removal of heat from the human body by means of radiation, conduction and evaporation, just as it is the prime requisite in the power plant. The efficiency of the human power plant is perhaps higher than that of other animals. There is perhaps a smaller percentage of waste heat as compared with useful energy.

Mr. Carrier discussed at some length the subject of air distribution and emphasized the necessity of research in this direction. He pointed out that although considerable valuable research had been done on many phases of air conditioning, information on proper air distribution was lacking, especially with respect to avoidance of drafts.

A considerable amount of important Chapter business was disposed of, including the election of officers for the coming year, as follows:

President, Russell Donnelly.
Vice-President, L. T. M. Ralston.
Secretary, W. A. Swain.
Treasurer, F. E. W. Beebe.
Board of Governors, A. L. Baum, Raymond Newcomb, A. J. Offner.

Western New York

April 13, 1931.—The April Meeting was held at the Hotel Buffalo, with an attendance of 52 members and guests.

After the various committee reports had been presented, President Davis introduced the speaker of the evening, C. J. Lundvall, field engineer for the National Tube Co., who rendered one of the most interesting lectures ever presented before this Chapter. Mr. Lundvall outlined the work done at the research laboratory maintained by the company and pointed out that through the cooperation of this laboratory with engineers, plumbing and heating contractors and plant managers, it is possible to develop materials which are suitable for the conditions to which they are subjected.

Mr. Lundvall then traced the manufacture of tubes, which he stated were, in their early stages, made of clay, lead, bronze and bamboo. He stated that the demand for pipe began with the commercial production of artificial gas in about 1815. Up to this time iron tubes had been made by hand, but shortly afterward machine-made butt welded pipe was developed in England, and between 1830 and 1834 the manufacture of machine-made pipe was begun in this country.

At the conclusion of Mr. Lundvall's talk he presented six reels of motion pictures, which showed the type of tubes used in ancient times by Greeks and Romans, the beginning of tube manufacture in America, ore mining, blast furnace operation, explanation of the butt-weld process and also the lap-weld process and the manufacture of couplings, all of which were very interesting.

The meeting was then opened to discussion which was thoroughly enjoyed.

Ontario

April 20, 1931.—The principal speaker of the evening was James Govan, consulting architect of Toronto, who gave a talk, which he illustrated with slides, on the subject of How Soon Will a Reduction of 60 Per Cent in Heating Plant Equipment Be Possible? Following Mr. Govan's talk there was a discussion on the various points which he brought out.

Pres. H. J. Church urged that as many members as possible attend the Semi-Annual Meeting of the Society to be held in Swampscott, Mass., June 22 to 25, 1931.

The annual election of officers was held, with the following results:

President—H. S. Moore.
Vice-President—J. S. Wood.
Secretary-Treasurer—H. R. Roth.
Board of Governors—H. J. Church, W. P. Boddington, M. W. Shears and J. S. Paterson.

This meeting was attended by 30 members and guests.

Pacific Northwest

April 6, 1931.—The April meeting, which was the largest in the history of this Chapter, was held at the Olympic Hotel in connection with the meeting of the *A. S. M. E.*

W. H. Carrier, President of the American Society of Heating and Ventilating Engineers, addressed the gathering on Servicing the Human Power Plant. His talk was illustrated by slides and was greatly appreciated by the members and guests.

Another of the guests, A. V. Hutchinson, National Secretary of the Society, spoke briefly on the activities of the Society. A rising vote of thanks was extended to both President Carrier and Secretary Hutchinson.

The regular order of business was taken care of, and the election of the officers to serve the Chapter for the ensuing year were announced as follows:

President—W. E. Beggs.
Vice-President—C. F. Twist.
Treasurer—W. W. Cox.
Secretary—Marius Anderson.
Board of Governors—E. O. Eastwood, E. L. Weber and W. Lyle Dudley.

Pittsburgh

April 6, 1931.—The regular monthly meeting of the Pittsburgh Chapter was held in the Assembly Room of the Fort Pitt Hotel. Forty-one members and guests were in attendance when the meeting was called to order by President Houghten.

F. A. Gunther, Chairman of the Program Committee, stated that so far no arrangements had been made for the next meeting, but that he had several very good prospects in view, any of which would furnish a profitable and enjoyable program for the last meeting of the year.

F. B. Orr, Treasurer, reported that he had followed instructions given him at the last meeting, and that the Chapter has a balance of $342.65 in its checking account.

Mr. Gunther introduced Miss Margaret Ingels, of the Carrier Engineering Corp., Newark, N. J., who gave a most interesting illustrated talk covering the principles and application of air conditioning for residences and public buildings.

After an interesting discussion of the subject, in which many participated, Miss Ingels was tendered a rising vote of thanks.

Heating and Piping Contractors Meeting in Louisville

Evidence of cooperative work in establishing standard heating and ventilating practice between the A. S. H. V. E. and the Heating and Piping Contractors National Association was clearly shown at the Association's Annual Convention in Louisville, Kentucky, May 6 to 9. Members of both Associations have taken prominent part in this work and who appeared on the program were Walter Klie, Cleveland, President of the Association; R. L. Spitzley, Detroit, President-Elect; H. M. Hart, Chicago, Chairman of the Special Committee on Boiler Output; R. S. Franklin, Boston, Chairman of the Committee on Standards, and A. C. Willard, Urbana, Professor of Heating and Ventilation, University of Illinois, who addressed the meeting on Research work on rooms heated by direct radiation.

In his President's address, Walter Klie classified those in the heating industry into three groups: The fatalists, the faddists, and between those two extremes, those who have a passion for facts and who base their action in any situation upon what the facts disclose. After describing these three groups, Mr. Klie made

eight definite recommendations which local associations and individual association members could follow:

1. Create more business by modernization and merchandising methods.
2. Definitely know and respect your costs, including overhead.
3. Endeavor to sell more contracts on quality instead of at the present destructive prices.
4. Be progressive in developing efficient high grade installations, keeping abreast with the most modern and economical methods.
5. Conduct the accounting department of your business in such a manner as to accurately reflect your costs and your business conditions.
6. Make it possible for the National Association to continue and improve these and other valuable services, and to be the medium for gathering all the relevant facts for the planning of sound readjustments, by increasing membership and prompt dues paying, and by patronizing, whenever possible, Bulletin advertisers to encourage continuation and increasing of such advertising.
7. Encourage the jobbers and manufacturers to stop the practice of carrying contractors who put their under-estimated costs and overhead into growing accounts payable, by paying your own bills promptly and encouraging those who do not discriminate against you, by keeping alive the habitual below labor and material cost bidder.
8. Give adequately of your time, your best constructive thoughts and your money to your Local and National Associations in these trying times, and develop a forward looking vision and a hopefulness born of faith in the fundamental soundness of our industry.

RAY L. SPITZLEY, DETROIT, MICHIGAN, PRESIDENT, HEATING AND PIPING CONTRACTORS NATIONAL ASSOCIATION

In closing, Mr. Klie paid a tribute to the cooperation which he has received from the National staff, headed by Mr. Fitts, the various committees, the Secretaries' Conference, and the Board of Directors; and closed by saying, "Follow these eight simple principles the coming year, beginning by active participation in this convention, and I assure you that progress will be made in the right direction."

Harry M. Hart, Chairman of the Sub-Committee on Boiler Output, reported an ever-increasing demand for the Association's Standards from consulting engineers, architects, and municipal officials. Mr. Hart stated that his Committee during the past year had interviewed representatives of 21 boiler manufacturers, and that the Standards of the Association now include recommendations on 3,782 different boilers. Other highlights of Mr. Hart's report follow:

During the fiscal year we have revised our ratings on steel boilers so that they now conform to the recommendations of the Steel Heating Boiler Institute for hand-fired boilers. The revisions and additions for our standards have necessitated approximately 25,000 separate computations.

The application of mechanical firing of heating boilers is becoming more popular all the time and boilers especially designed for such firing are coming on the market in ever increasing quantities. This, of course, gives your committee a lot of work, because as soon as these new devices come on the market we are flooded with requests for our recommendations on them.

Until the Oil Burner Association and the Stoker Association adopt some definite standards governing the installation of their products, the question of boiler outputs and efficiencies, when mechanically fired, will remain somewhat uncertain. Therefore, your committee feels that no change from our recommendations for hand-fired boilers should be made when boilers are fired by mechanical stoker, oil or gas.

When boilers are designed for mechanical firing only, we demand that the manufacturers furnish us complete performance charts made from tests conducted by qualified engineers before we make any recommendations to our members. This will no doubt explain why so few of such boilers appear in the Association's Book of Standards at this time.

Ralph S. Franklin, Chairman of the Committee on Standards, reported that during the year a revised edition of the Engineering Standards had been distributed to the membership without cost, and as far as the Committee could ascertain had been very favorably received. Other projects reported on by Mr. Franklin were the establishment of a standard method of determining the proper amount of radiation for any type of structure; the determination of hot water pipe sizes for gravity circulating systems; the completion of arrangements with the Engineering Department of New York University for a long series of tests on a complete, isolated building to determine the effect of varying weather conditions on the steam consumption of the entire building and of each separate room of the building, and an extended series of tests to secure more accurate data on the heat losses through concrete floors on the ground.

In reporting on one of these major activities, Mr. Franklin said:

Perhaps the most important and, to the members, the most interesting part of the Committee's work during the past year is the result of its studies on the determination of hot water pipe sizes for gravity circulating systems.

In view of the fact that this type of data must necessarily be applicable to installations with entirely different conditions, your Committee became convinced that any standard submitted must provide for orifice control.

The most recent and comprehensive data on this phase of hot water circulation were presented before the American Society of Heating and Ventilating Engineers in a paper by E. G. Smith and F. E. Giesecke of the Engineering Experiment Station of the Agricultural and Mechanical College of Texas, and these data were used as the basis for the proposed standard.

The principles therein laid down have been applied in a number of actual installations by members of this Committee, and have given uniformly satisfactory results. They have also been applied on unsatisfactory old installations and have in each case been the means of correcting the trouble.

It is true that the use of orifices in this type of work is not new. Many heating contractors have made use of orifices at various times for a great many years, but the determination of size and points of application has usually been by the "cut and try" method. To the best of the Committee's knowledge, this method is the first which requires an orifice at each radiator and determines its size before installation.

Professor Willard gave a very interesting illustrated talk on heating rooms with radiators. His slides showed the effect of enclosures of different types on the condensation of steam by the radiator and also on the ceiling temperatures, breathing line temperature, and the temperature of lower levels. His talk and his slides demonstrated the necessity of the heating engineer and the installing contractor taking definite notice of the type of enclosure used, as a properly designed enclosure will increase the effectiveness of the radiator and save steam while the improperly designed enclosure will simply lower the temperature of the entire room as though sections had been cut off the radiator. He used other slides to demonstrate the value of the test room at the University of Illinois, showing the equipment available for testing and also some very interesting results obtained in studying single and double glazed windows and the fall of temperature through walls in still air and with winds of different velocities.

Many other reports were made on other activities of the Asso-

ciation for the past year and approved by the membership. The confidence with which the various phases of the Association's activities in standardization, trade promotion, Certified Heating, welding, and merchandising have been so successfully promoted and the general interest in these activities by the entire membership indicates that the heating industry feels that the time is ripe for constructive work in preparation for the general upturn in business expected later in this year.

At the closing session of the convention, Ray L. Spitzley, who has been active in Association activities for a number of years was unanimously elected to the Presidency of the Association for the coming year.

The officers and membership of the A. S. H. V. E. congratulate their fellow member and wish for him a successful administration and for the heating industry prosperity and profit for 1931.

Universal Atlas Handbook of Concrete Construction

Useful and up-to-date information on concrete, both plain and reinforced, is contained in the new *Handbook of Concrete Construction*, published by the Universal Atlas Cement Co., Chicago.

The book is written from a practical rather than a technical standpoint. It makes no attempt to cover the field of large construction projects requiring skilled technical supervision. Instead, it is a handy pocket manual of general information on concrete, applying particularly to smaller structures.

Beginning with a concise discussion on concrete as a construction material, the first chapter is devoted to material on how to make good concrete. It traces, step by step, different methods of mixing, placing and curing concrete on the average job. This includes consideration of materials, equipment and labor and an easily understood presentation of the water-cement ratio method. In the second chapter the basic rules covering the reinforcing of concrete are briefly presented. The remaining three chapters discuss forms of concrete, construction of small concrete structures and simple rules for estimating quantities and costs.

The book has 208 pages, is well illustrated, and contains 48 accurate, easily read tables on quantities, sizes, loads and related subjects. It has a convenient index by means of which the reader can readily refer to any part of the book desired.

A. S. R. E. Western Meeting

The 18th Western Meeting of the American Society of Refrigerating Engineers was held in Kansas City, Mo., May 6, 7 and 8, 1931, with headquarters at the Kansas City Athletic Club.

The technical sessions were held in the morning, and were open to the public. The papers presented, which covered the latest problems in refrigeration, included the following:

Paper as a Refrigerator Insulation, J. L. Knight, General Electric Co., Kansas City, Mo.

Refrigeration in Air Conditioning, Maurice Olchoff, Natkin Engineering Co., Kansas City, Mo.

Insulated Cold Rooms Used in Various Industries, H. J. Krampe, Armstrong Cork Co., Lancaster, Pa.

A very interesting social program was enjoyed by all, and the meeting was declared to be one of the most interesting and successful western meetings of this Society.

American Construction Council Meeting

The Eighth Annual Spring Meeting of the *American Construction Council* was held in New York, May 11, 1931. This meeting was devoted to the discussion of two of the most important subjects now confronting business in relation to construction activities, namely (1) building financing and its relation to public welfare, and (2) the place of quasi-public works, including slum and tenement clearance in the stabilization of business.

Standards for Pipe Flanges and Fittings

The American tentative standard for cast-iron pipe flanges and flanged fittings has recently been approved by the *American Standards Association*. This standard is for a maximum working steam pressure of 25 lb gage.

The sponsor organizations include the *Heating and Piping Contractors National Association, Manufacturers' Standardization Society of Valve and Fittings Industry*, and *The American Society of Mechanical Engineers*.

Yale Institute of Human Relations Dedicated

The Institute of Human Relations at Yale University, New Haven, Conn., which was established two years ago for a scientific investigation of man's behavior from the physical, mental and social viewpoints, was dedicated and opened on May 9, 1931. The function of the institute is to stimulate research as well as to promote the integration of knowledge bearing upon problems of human well-being.

The research activities conducted directly by or in association with the institute fall roughly into three main groups. In the first group belong studies designed to bring out the basic facts concerning the human and social organism; in the second are those dealing with immediate problems of human welfare, such as unemployment, crime and public health, all of which demand the cooperation of specialists in many fields, and in the third group are studies dealing with specific problems in such fields as law, medicine, religion, education, economics and government.

American Oil Burner Association Closes Successful Convention

The eighth annual convention and show of the *American Oil Burner Association* which was held in Philadelphia from April 13 to 18 is believed by those who attended to have been the most successful event ever held by this Association. The eighty-four exhibitors who took part in the annual oil burner show constitute a record number and the member registration and attendance at the two merchandising and one engineering sessions were greater than at any previous convention.

Attendance at the show also was greater than in any previous year with registrations totalling approximately 12,000. This is 2,000 more than the number of visitors who attended the 1930 show in Chicago. It was estimated that approximately 1,000 of the 12,000 visitors at Philadelphia were members of the Association, and 2,500 were non-members who were identified in some way with the oil burner industry. The remaining 8,500 were the general public including home owners and builders.

The annual meeting for members resulted in the election of two new members to the Board of Directors and the re-election of eight others whose terms expired this year. J. P. McCarthy, New York, and Charles T. Bendix, Freeport, Long Island, were the newly elected directors. Both will serve for three years. Mr. McCarthy succeeds Meade Durbrow and Mr. Bendix succeeds John W. Scott. The new board met for the first time on April 16 and re-elected Walter F. Tant, of Detroit, president. It is Mr. Tant's second term.

The convention was particularly significant for dealers and resulted in the adoption of plans for an active and aggressive dealer organization as a division of the *American Oil Burner Association*. Among the more important duties of the dealer division will be the strengthening of existing local associations and the formation of new ones; the correction of obsolete ordinances; distribution of engineering data; setting up of cost recording systems and advice on business management problems; adoption of a code of ethics; market research work; cooperative advertising; assistance to dealers in meeting competition and other activities. The plan will be put into operation as soon as the head of the division is selected.

CANDIDATES FOR MEMBERSHIP

The Constitution of the Society, as now amended, requires the following mode of procedure in voting on applicants for membership in the Society. All applications for membership are to be sent to the Secretary and the names of applicants and their references shall be printed in the next issue of the Journal of the Society or sent to the members in other approved manner as ordered by the Council. When replies are received from references, the Candidate's application shall be submitted to and acted upon by the Membership Committee as soon as possible.

When the Membership Committee has acted favorably upon a Candidate's application and assigned his grade, the Council shall vote upon the election of the proposed Candidate for membership by letter ballot. During the past month 13 applications for membership have been received and the names of these men and their sponsors are published in the following list.

Members are requested to scrutinize the list with care. The Membership Committee, and in turn the Council, urge the members to assume their share of the responsibility of receiving these candidates into membership by advising the Secretary promptly of any whose eligibility for membership is in any way questioned.

All correspondence in regard to such matters is strictly confidential, and is solely for the good of the Society, which it is the duty of every member to promote.

Unless objection is made by some members by June 15, 1931, these candidates will be balloted upon by the Council. Those elected to membership will be notified by the Secretary immediately after election.

CANDIDATES	REFERENCES	
	Proposers	*Seconders*
BLITZ, EMMANUEL, Engr., Andrew J. Thomas, Archt., New York, N. Y.	A. J. Offner S. A. S. Patorno	O. E. Goldschmidt L. T. M. Ralston
EDWARDS, DON CALVIN, JR., Sales Engr., Carrier-York Corp., Nashville, Tenn.	Thornton Lewis H. P. Gant	D. E. French C. D. Graham
GIFFEN, J. KERR, Stanton Heater Co., Martins Ferry, Ohio.		
GROSSMAN, HARRY A., Owner, H. A. Grossman Co., St. Louis, Mo.	J. W. Cooper R. M. Rosebrough	C. R. Davis S. G. Hallett
KEEFE, EDMUND T., President, Underground Steam Construction Co., Boston, Mass.	J. F. Tuttle D. S. Boyden	W. T. Jones W. P. Mower
KELLY, JOSEPH A., Asst. Chief Engineer, John J. Nesbitt, Inc., Philadelphia, Pa.	G. H. Stoever W. F. Bilyeu	M. F. Blankin L. C. Davidson
KIRKPATRICK, ARTHUR H., Specialty Engr., Ilg Electric Ventilating Co., Chicago, Ill.	J. M. Frank A. G. Sutcliffe	M. K. Arenberg B. L. Casey
MATHIS, GEORGE ANTHONY, Sales Manager, New York Blower Co., Cleveland, Ohio.	M. F. Rather J. E. Beyer	S. H. Givelber
MCELGIN, JOHN WILBERT, Asst. to Experimental Engineer, John J. Nesbitt, Inc., Philadelphia, Pa.	A. J. Nesbitt W. F. Bilyeu	M. F. Blankin L. C. Davidson
MYERS, EDDIE VERN, Vice-Pres. & Supt., A. S. Hunter & Son, Inc., Willard, Ohio.	R. G. Davis F. H. Morris	Frederick Leonhard *(Non-Member)* C. F. Eveleth
POWERS, EDGAR CHESTER, Sales Engr., Rome Radiation Co., Philadelphia, Pa.	R. E. Jones H. G. Black	C. B. Eastman F. H. Gawthrop
ROSEBERRY, JOHN HENRY, Manufacturer's Agent, 1211 Genessee Bldg., Buffalo, N. Y.	M. C. Beman F. H. Burke	Joseph Davis J. J. Landers
THOMAS, EARL E., General Electric Co., Los Angeles, Calif.	O. W. Ott A. H. Simonds	Leo Hungerford C. S. Anderson

Candidates Elected

In past issues of the Journal of the Society the names of the following men were listed as Candidates for Membership. The membership grade of each Candidate has been assigned by the Membership Committee and balloted upon by the Council. We are now instructed by the Council to post herewith, as required by Art. II, Sec. VIII, of the By-Laws, the following list of candidates elected:

MEMBERS

DUBOIS, LOUIS J., York Ice Machinery Corp., York, Penna.

MAY, EDWARD M., Superintendent, Combustioneer, Inc., Goshen, Ind.

REDSTONE, ARTHUR L., Research Engineer, Proctor & Schwartz, Philadelphia, Penna.

WHALEY, RALPH S., President, Power Plant Engrg. Co., Seattle, Wash.

ASSOCIATES

CROSS, ROBERT EARL, Sales Engr., Minneapolis-Honeywell Regulator Co., Boston, Mass.

NEWMAN, CHARLES THOMAS, Owner of Business, C. T. Newman, Los Angeles, Calif.

SMITH, GEORGE G., Board of Education, Chicago, Ill.

JUNIORS

CHRISTMAN, WILLIAM F., Theatre Air Conditioning Engineer, Carrier Engineering Corp., New York, N. Y.

HOLLISTER, EDMUND WALLACE, Field Supt., New York University, New York, N. Y.

STERNBERG, EDWIN, Designing Engr., Carrier Engrg. Corp., New York, N. Y.

WOOD, FREDERICK C., Sales Engr., York Ice Machinery Corp., York, Penna.

Heating · Piping and Air Conditioning EDITORIALS

Misleading Publicity a Menace

"After all, air conditioning is merely the providing of suitable temperature and humidity to the air." This statement appeared in a syndicated news release and is typical of a great deal of newspaper publicity that is published and which newspaper editors use to capture the popular fancy and to intrigue the imagination of their readers. A few days after the appearance of this statement, an encyclopedia salesman called at a certain engineering office and wanted to know how he could get into the air conditioning business.

In some cases, it is a happy situation for the engineer when newspapers the country over publish articles about his particular business. But human enthusiasm is not provided with a governor or regulator as are many types of engineering equipment, so newspaper writers, finding ready consumption of their pseudo-engineering articles on air conditioning, sometimes draw from their imaginations for further copy. The result is that much unfortunate and exaggerated publicity has been given to air conditioning and the public often is led to expect more than can be attained by any engineering knowledge we now possess or any equipment that has been devised. For instance, a woman went into an office of a heating engineer recently and wanted to know about "This new system of heating by radio." She had read about it in the newspapers.

Uncensored newspaper publicity has brought into the field a new species of engineer. He thinks that all he has to do is to provide heat, moisture, motion and cleanliness. About the amounts, velocities, distribution, and actual exact means of providing comfort for occupants of buildings or suitable air conditions for manufactured products he knows little. He picks up a few engineering terms and endeavors to capitalize on the publicity air conditioning has received. In some instances, he designs equipment which also is based on meager engineering knowledge.

Installations of this kind are encountered frequently by skilled air conditioning engineers. True, this equipment often finds its way into the scrap heap in a year or so but in the meantime air conditioning has been discredited and some years may elapse before the owner of the building or factory in which the inadequate equipment was installed comes around to the knowledge that there is air conditioning of a different kind.

This tendency in our commercial affairs has a more far-reaching effect than is at once apparent. It affects not only air conditioning but many other industries as well. High-pressure salesmanship lacking accurate technical knowledge is a menace to many industries and is charged with the responsibility for a large part of our present business depression. When, by means of high-pressure salesmanship lacking technical knowledge, a large number of people are over-sold or sold more than they really want to buy or should buy, the effect can be felt in the business world. When groups of people are sold equipment which does not function as it is supposed to function, the natural reaction is to keep money closely guarded and to refuse to release it for any purpose not absolutely necessary. Such salesmanship and such engineering create a lack of confidence in our entire business structure.

It is impossible, of course, to make good engineers of all those who suddenly decide that there is money waiting for them in the air conditioning field, but there are ways of combating this unwise publicity. One way is by the publication of concise, exact air conditioning articles such as appear in this journal and which say by implication, "This is a job for a skilled engineer and not for the man whose chief ability is the use of engineering phrases he does not understand."

Three Standard Formulas

The other day, an engineer worked a problem three times, using a different formula in each case. Each has been widely accepted and could be called a "standard" formula. The lowest answer obtained was about 35 per cent under the highest; the other answer was approximately midway between the two extremes. Which is the correct solution is naturally a puzzling and somewhat disconcerting question.

The engineer must maintain an open mind, even where formulas and methods that have been used—with a degree of success, at that—for several years are concerned. The methods of research are becoming more accurate all the time. Conditions uncontrollable a decade (plus or minus) past are now carefully considered when experimental results are analyzed. The necessary measuring instruments are, in the majority of instances, refined to greater degrees.

Formulas several years old are not necessarily inaccurate, of course. But where better research methods have replaced the less refined procedures of other days,

and in cases where two or more fairly standard formulas give widely differing results, it seems advisable to scrutinize with care the original data, rather than to accept without question experimental results just because they have been accepted up to the present.

Research work based on an erroneous assumption can be no more accurate than was the basis. It is of great importance that the engineer can depend on the accuracy of the data he uses. Fortunately, no one realizes this more than does the research engineer.

More Clothes or Less

Most engineers regard 66 or 68 F with suitable humidity the ideal temperature for winter. In a certain large commercial establishment, the temperature during the cold seasons is maintained at 70 F. In spite of this, a large number of the women employes complain of being chilly while some of the men work with their coats off. In the same establishment during the warm days of summer, no complaints are heard from the women but the amount of work done by the men is influenced by the heat. It would appear that the problem of room temperature for those heavily and lightly clothed is not entirely solved.

But there are many other kinds of people in their heat requirements. There may be in the same room not only those that are heavily and those that are lightly clothed, but also those that are sick and those that are well, those that are used to Northern climates and those that are used to the temperate breezes of the South, those that are engaged in sedentary occupations and those that gain their livelihood by active work, those that are young and those that are old, those whose basal metabolisms produce 280 Btu per hour and those who naturally generate 350 Btu per hour. It is the problem of the heating and air conditioning engineer to keep them all comfortable and happy.

One health organization states that clothing worn by women in this country ranges from 1 lb. 6 oz. to 3½ lb. The former, including shoes, is the weight of the complete attire of the modern girl of 18 and the latter the clothing worn by an old-fashioned woman of 61. The average weight was 2 lb. 10 oz. The weight of men's clothing varies from 6¾ lb. to 10¼ lb.—with an average of 8 lb. 6 oz. or more than three times that of women's clothing.

It is, of course, difficult to effect any radical change in dress of men or women. The average man will accept without question standard customs in dress that have no pretense to beauty or hygienic consistency because in doing so he does not make himself conspicuous. During the snappiest cold spells of winter, women will wear stockings no thicker than they wear in July or August.

But one who has observed these conditions may be permitted to present a solution even though human nature may prevent any extended application of it. It is obvious that men are over-clothed in summer. Are they also over-clothed in winter? Should winter room temperatures be adjusted to the scantily clothed female or the heavily upholstered male? Should summer cooling take into consideration the heat transmission through 2 lb. 10 oz. of clothing or through 8 lb. 6 oz? Here lies the problem. If men are over-clothed in summer, is it possible that women are under-clothed in winter? Is their 1 lb. 6 oz. of clothing sufficient to keep the female in good health during the cold weather? It might be if our skins could be relied upon as effective heat regulators under all conditions. But our civilized skins are not trained to such temperature changes as they encounter in our modern manner of living and at such times clothing is necessary to conserve heat and health.

This is borne out to some extent by the report from a large insurance company which states that in industry women show a higher rate of respiratory affections and that for some time it has been apparent from the mortality statistics that the reduction in death rate from tuberculosis has not been so favorable for young women as for young men in the same age group.

So it would seem the part of common sense to make up for our inadequate skin functions to some extent by proper clothing. It is apparent that with some common sense about clothing, our cold and grippe epidemics would not be the menacing factors that they are; it would also save the heating and air conditioning engineer a lot of worry.

Don't Miss These!

Interesting data on the initial and operating costs of air conditioning systems with and without refrigeration are included in I. C. Baker's paper on air conditioning press rooms, which is to appear next month. In this article, the author shows what air conditioning can do in improving production and decreasing costs in printing plants and discusses the methods of doing it. Industrial air conditioners will find it an article to be studied with care.

It behooves the wide-awake industrialist to prepare for the replacement of his present plant long before its apparent usefulness has been outlived, says E. H. Barry in his article on high pressure steam and the modern industrial plant to be published in the July HEATING, PIPING AND AIR CONDITIONING. Mr. Barry's discussion is particularly timely, with the wide interest in higher steam pressures for industrial uses these days.

"Open for Discussion"

By-Product Generation of Electricity in District Heating Plants. Welded Piping

Paris District Heating System Handicapped

By A. C. Coonradt*

TO AN engineer, familiar with American conditions and practice, the Paris district heating project would seem to be an unfortunate undertaking.[1] Unfortunate, because central station heating can add so much to the comfort and convenience of city life and at the same time offer a legitimate field for the profitable development of a relatively new public utility, if given an opportunity at the start to build up under favorable conditions.

The conditions under which the Compagnie Parisienne de Chauffage Urbain is starting seem decidedly unfavorable and, to a great extent, unnecessarily so.

The city ban on boiler plants would seem to defeat its own end. The object of prohibiting the operation of boilers within the city limits is evidently to reduce smoke and dirt, fire and accident hazards, and traffic congestion. As regards manufacturing plants and electric plants, their stand is probably well taken and works no particular hardship on the operating companies, but central station heating presents an altogether different situation. In this case one boiler plant replaces many smaller ones scattered over a large area. The central station plant is designed by skilled engineers for high combustion efficiency and, where modern apparatus is installed, practically no dirt or soot is emitted. Fire and accident hazards are reduced through the concentration of the many fires in one place under expert attendance. The delivery of fuel to one spot, chosen in relation to rail or water transportation facilities will cause no increase in the traffic congestion and should eliminate much of the dirt and noise resulting from the delivery of fuel over a wide area. That the inferences made here are borne out in practice is shown by the tests made in New York City. A study of the results of these tests make it strikingly apparent that the small fires in the heating plants of some of the apartment houses, the residences, and the tugs and ferries on the rivers are the principal contributors to atmospheric pollution.

Load Density

While the matter of plant location appears to be the most seriously adverse of the factors against which this company must contend, that of load density is also likely to prove disappointing. If one may judge the entire district by the photograph which shows the line in Rue de Bercy, the complete absence of large buildings would indicate that the amount of steam required for heating per unit length of steam main will be relatively low. This, of course, means that high rates will have to be charged if service is to be furnished at a profit.

The pipe line construction appears to be very good—perhaps too good when one considers that the fixed charges on the investment in the line continue throughout the entire year while the operating charges predominate only during the heating season. The ad-

VIEW OF LA RAPEE LINE IN RUE DE BERCY, GARE DE LYON DISTRICT, SHOWING PIPES IN PLACE BUT NOT INSULATED

* College of Engineering, New York University.

[1] *Steam Piping for New District Heating System in Paris*, by Philippe Schereschewsky. HEATING, PIPING AND AIR CONDITIONING, November, 1930, p. 914.

visability of piping the returns back to the plant is open to question. With a line as long as this it would seem, offhand, that modern methods of feed water treatment of makeup water at the plant would prove less costly. The best way of handling this situation would of course be determined by an engineering analysis and no doubt such an analysis was made by the designers of the system.

Bled Steam

It is mentioned that the intention is to use exhaust steam or steam bled from the turbo-generators operated by the electric company. While there are a few central heating systems in this country operating in this way, many companies have given up the idea and are operating with steam direct from the boilers.

The advantages to be gained by the combination operation are often more fancied than real. Unless extremely high initial boiler pressure is used, either a very small pressure drop is obtained through the turbine, resulting in low output, or the pressure for the steam main is very low. A low steam main pressure requires a large pipe and a high investment in pipe line. As the amount of load on the pipe line increases, the pipe line pressure at the plant must be increased in order to maintain a required pressure at the end of the pipe line. This increase in pipe line pressure means a higher back pressure and a lower electrical capacity for the turbine. This coupled with the fact that heating and electric loads are not coincident in intensity, sometimes makes the combination operation unprofitable.—*A. C. C.*

By-Product Generation of Power

The interesting comments of Mr. Coonradt have been transmitted to M. Schereschewsky but time has not permitted a reply by him in this issue. Without presuming to speak for M. Schereschewsky the writer offers the following observations.

To understand the prohibition that the city of Paris has placed upon the construction of boiler plants within its limits, one must bear in mind that the city authorities exert control over the character and design of buildings to a degree unheard of in America. Undoubtedly this prohibition, making necessary the transmission of steam over the long distance from the power plant to the heating district, imposed a severe burden upon the enterprise.

The same influence is reflected in the density of the heating load, which, as Mr. Coonradt points out, is quite low when judged by American standards. The low density is due largely to governmental restrictions upon the height to which buildings may be built.

Regarding the byproduct generation of electricity, the underlying reasons for its lack of popularity in America are (a) the comparatively low and decreasing cost of producing electricity in condensing plants, which cost fixes the value of any byproduct current which might be generated, and (b) the relation between fuel costs and investment costs. Evaluation of these factors for the conditions prevailing in France may obviously lead to quite different conclusions.

Inasmuch as the steam for heating is being taken from an electrical generating station in which turbines are already installed, it would seem quite logical to bleed the necessary steam from those turbines as described by M. Schereschewsky rather than to supply live steam direct from the boilers.—*J. H. Walker,* The Detroit Edison Company.

Welded Piping

By G. K. Saurwein*

HIGHER pressures and temperatures as they affect piping were ably discussed by Mr. Harman in his recent paper.[1] The part of particular interest to the writer is that on welded joints for pipe lines which perhaps might have been developed a little further.

Welding has come out of the experimental stage. During the last ten years it has been extensively used on pipe lines and on piping systems within buildings. Of late, a number of power plants operating on pressures up to four hundred pounds have been constructed with welded piping where all joints have been welded except those adjacent to valves, pressure reducers and other specialties which might require occasional replacement. One twelve-hundred-pound plant also had its piping made up with welded joints.

Power Plant Recently Completed

A power plant just completed under the writer's supervision is fitted with welded piping throughout, except for valve and specialty flanges. This plant, operating at two hundred and fifty pounds pressure and one hundred degrees superheat, is giving entire satisfaction. The piping is light in weight, presents a neat appearance, and joint troubles are eliminated.

Changes Made Easily

The writer has found it inexpensive to make changes by means of the welding torch. The operation of burning a hole in the side of a pipe and attaching a welded nozzle consumes little time and materials; the welded joint in the line has been trouble-free and inexpensive to weld. The reason given by some engineers for not welding valves into the line is that the heating might distort the valve body and thus destroy the tightness of the valve. But valves with welding ends sufficiently long so that no trouble will be had from distortion can be used in many locations.

*Engineer, Harvard University.
[1] Higher Pressures and Temperatures as They Affect the Design, Installation and Maintenance of Piping, by J. J. Harman. Heating, Piping and Air Conditioning, Dec. 1930 to Feb. 1931.

Maintenance of Air Washers

By Samuel R. Lewis*

I WAS interested in William A. Hanley's "Maintenance of Air Conditioning Equipment," published recently.[1] In order to reduce the excessive renewal-cost of air conditioning equipment, some very successful installations have been made using non-metallic materials. Since air washers using such material are *assembled* devices they may be designed by the plant engineer or by the consulting engineer.

These structures may have concrete tanks, and

*Consulting engineer, Chicago, Ill.
[1] Page 905, November, 1930, Heating, Piping and Air Conditioning.

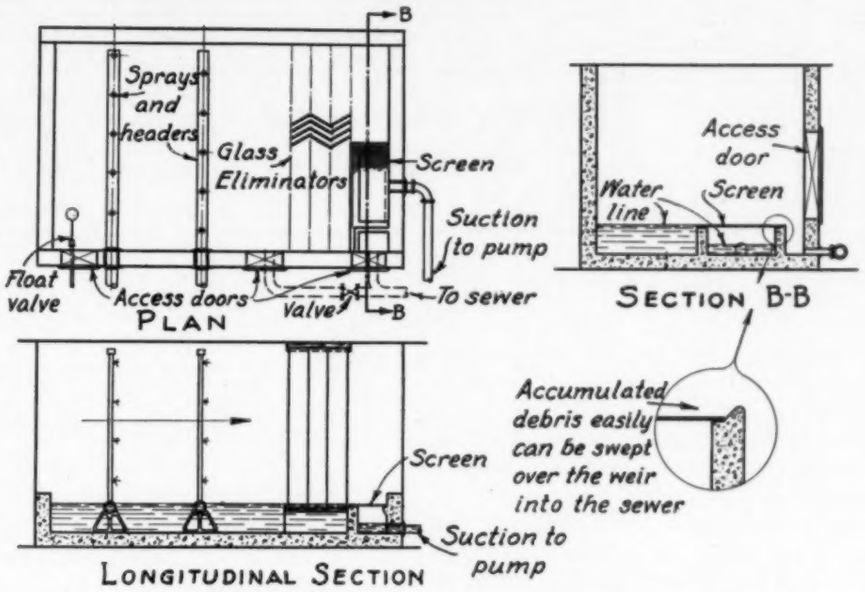


Diagram of Air Washer, Showing a Detail of the Screen Arrangement to Catch Debris

may use asbestos-portland cement housings, though when the concrete, as it easily may, extends well above the water line, the housings will endure fairly well when made of galvanized iron. We have used glazed tile, and glass and slate. The eliminator plates are made of ribbed wire glass, with the ribs facing toward the spray, so that each plate is like a washboard presented to the "air-water-mist" flow. The glass plates rest in grooved soft wood planks; this material, especially if creosote-treated, being very durable for the top of the glass, while any wood, if kept wet all of the time as is that at the bottom of the glass, will last indefinitely.

Corroborative experience is as follows: I have had occasion to replace a number of wooden dams, built thirty years ago by my father, in northern Wisconsin. Above the water line, where sun and rain and snow and frost can work their potent evil, the native wood splinters and rots and must be replaced at intervals of about five years. Below the water line the original jack pine and tamarack planks are as bright colored and as firm as when they were installed. The spikes, however, which held the old planks are corroded clear through and the failure of the spikes is the basis of necessity for underwater renewal.

We have designed and caused to be built, about twelve such air washers. All of them have fulfilled their functions, with low maintenance.

Horizontal baffles in air washers often have not proved satisfactory, since the down-falling streams of water cannot wash off the accretions of mud.

Screens for separating out the pigeon feathers, cigar butts, moths, leaves, etc., which collect in city air washer tanks and which would clog the sprayheads, are of many types. Many of them are inaccessible for cleaning and many of them never are cleaned.

One suggestion, in successful operation, is to have the screen a very substantial perforated copper plate placed horizontally at the water line of the washer tank. Only such debris as will float thus can come to the screen. There is an access door directly in front of the screen, and arrangements are made so that any one who will use a floor brush can reach in through the door and swab off the top of the screen from outside, very easily.

Water-tight Roof Gives Protection

When an air washer is placed in any building, an important point is first to build a water-tight roof, having a drain under the entire zone occupied by the device with all of its piping. Without such protection, when the eventual leak occurs, due to gland or gasket or due to a spray head gone wild, or to corrosion, there will be embarrassment for whoever and whatever there may be underneath!

Since fans can be operated, and often are operated, without running the air washer pumps, the duct systems on the discharge side of the air washers may become very dusty.

If the water in the washer for any reason is not kept cooler than the air in summer, much discomfort may follow the use of the washer, and following such non-operation the ducts will get dusty.

One concludes that air cleaning and the functions of an ordinary air washer, which primarily is a heat-transfer agent, are separate, and usually are incompatible. One can not hope to achieve the best results in both air cleaning and heat-transfer from one device. It is wise, therefore, to use an air filter, ahead of the air washer. Under such conditions the air washer serves as a humidifier and dehumidifier and cooler, and may even serve as a heater, and may be used as required to meet seasonal changes without prejudice to the dust effect on the building.

PRACTICAL PIPING PROBLEMS

How a Vacuum Return Heating System was Inspected and Improved

by **W. H. WILSON**

Testing Heating System Return Lines

The vacuum return system of steam heating is widely used and to obtain its benefits, it is necessary to keep the return pipe lines free from leakage and the steam traps in proper working condition.

The accompanying diagram shows the location of the supply and return lines for a portion of an extensive heating system. Building *A* is located a thousand feet from the vacuum pump, shown at *K* in building *C*. The heating return line *E* from building *A* is located in an underground tunnel, passing through building *B*. The steam heating supply lines are shown at *D*.

Building *A* is a three-story building, with approximately a total of twelve thousand square feet of direct radiation.

The heating system in this building, when first installed, gave satisfactory results for a number of years before trouble was experienced in extreme cold weather, especially in the section of the building shown at *N*. Complaints received on account of the lack of proper heating were taken care of by raising the steam pressure.

A change in the operating force brought out different ideas as to rectifying this trouble and it was decided to install an addition to the steam supply lines by connecting across from building *B* to building *A*, shown at *R*. The use of this connecting pipe line on the steam supply showed improvement in the heating, but at times trouble was experienced in building *A* at the location *O*.

After two heating seasons and with another change in the operating force, the question of heating this building was again considered. It was then decided to give the heating return lines an inspection and test. The greater portion of the return pipes were located in tunnels, for the most part not readily accessible. Some of the branch lines were under floors, others under concrete walks.

Thorough Test Made

A thorough test was made. Sections of the line were disconnected, plugged and compressed air was used for testing. Places were found where the pipe was worn away by corrosion. Fig. 2 shows a sample of the pipe that required renewing. This pipe was in one of the tunnels, covered with dirt and refuse material.

When the repair work was completed a test was made by using compressed air pressure to determine that the return lines were tight enough for practical use.

Traps Tested

The steam traps were then inspected and a number were found to be leaking and passing steam. Two high pressure steam traps were found to be leaking badly. These traps were put in proper order and the discharge piped to another point to keep them clear of the return lines.

Stop Cock Worn

A cold water line, used for process work and located in the attic of the building, was found to have a circulation connection to prevent this line from freezing. The circulation was connected directly into the heating return lines. The amount of water passing through this circulation connection was controlled by a stop cock of the core type.

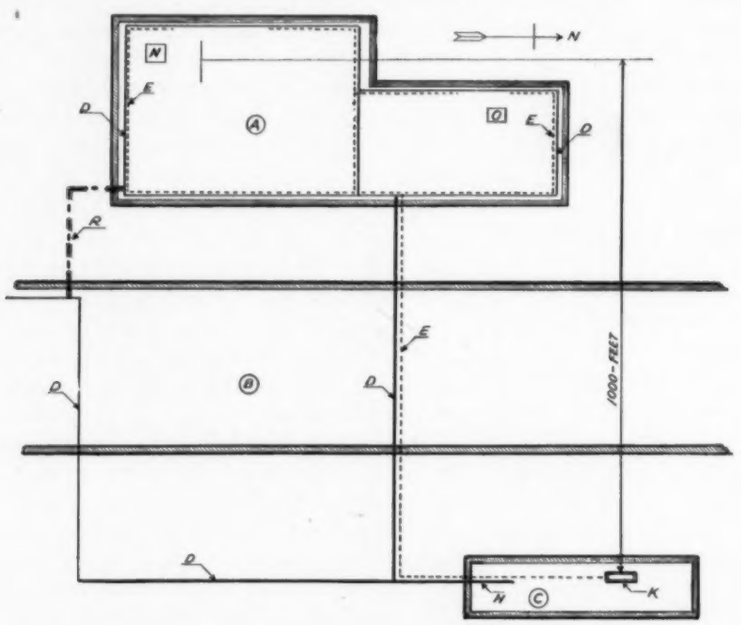


FIG. 1 — DIAGRAM SHOWING LOCATION OF SUPPLY AND RETURN LINES

When first installed the stop cock was probably adjusted so as to permit only a small amount of water to pass into the return line. The action of the water in the restricted opening had worn the core and body of the stop cock so that enough water was passing into the return line to retard the condensation from some of the heating system radiators. Fig. 3 shows the condition of the stop cock when it was removed.

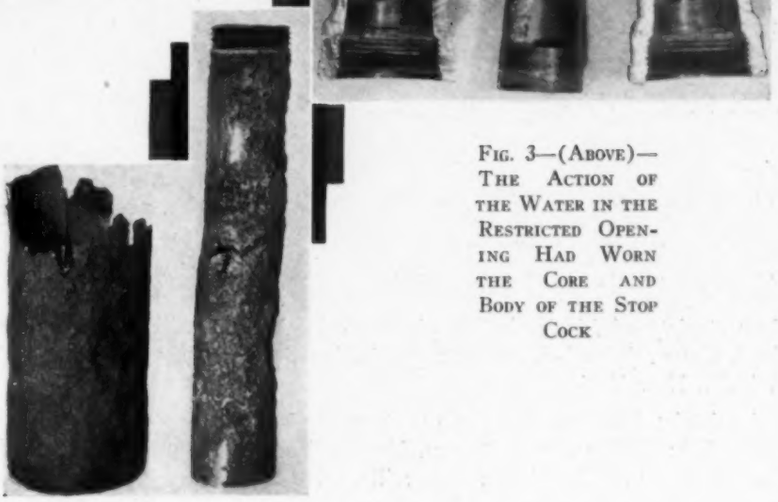


FIG. 3—(ABOVE)—THE ACTION OF THE WATER IN THE RESTRICTED OPENING HAD WORN THE CORE AND BODY OF THE STOP COCK

FIG. 2—PLACES WERE FOUND WHERE THE PIPE WAS WORN AWAY

After the repairs mentioned were made the heating system was put in service. A vacuum of from two to three inches was maintained on the lines at the point farthest from the vacuum pump. No trouble was experienced in heating, with a low steam pressure, varying from ¼- to ½-lb. in mild weather, to 1- or 1½-lb. in cold weather, the northerly and westerly winds having a bearing on the heating.

The connecting supply line *R* is still used and it has been noticed that it helps in getting the radiators heated quicker when the steam is first turned on in the building. It assists in heating when the steam pressure is very low in extremely cold weather. In order to verify that the real cause of the trouble, in this case, was the defective return lines and bad condition of the traps, this connecting supply line has been shut off for several days at a time, in cold weather, without any complaints being received regarding the heat from the occupants of the building.

Vacuum Gages an Aid

It should be a simple matter to determine when the vacuum return lines are not in proper working order. This important matter is frequently overlooked. Vacuum gages may be installed at the remote points of the return lines, distant from the vacuum pump, in convenient locations, so that the amount of vacuum on the pipe lines can be noted at a glance during the regular maintenance inspections.

A handy article for the maintenance man is the cap of a thermostatic trap with the thermostatic element removed, a ¼-in. tapping in the cap, with a vacuum gage or a compound vacuum and pressure gage inserted in the top of the cap. When it is desired to note the amount of vacuum at any particular location, shut off a radiator, remove the thermostatic top of the radiator trap and apply the cap with the gage. After the gage reading has been noted, remove the test cap and gage, replace the thermostatic top of the trap and put the radiator in service.

To keep the return lines and traps of a large heating system, covering an extensive territory, in proper working condition requires close inspection, frequent testing and a generous amount of work.

A. S. T. M. Sectional Committee on Cast-Iron Pipe

The sectional committee on specifications for cast-iron pipe of the A. S. T. M. has completed, during the year 1930, considerable work on programs thus far adopted for tests on cast-iron pipe and fittings. A considerable portion of the work was done at the universities mentioned below.

The work at the University of Illinois, under the direction of M. L. Enger, involved strength tests on certain sizes of pit-cast pipe and on 6 and 12-in. tees, crosses, and wyes. Results show that slight changes in the dimensions at important points will contribute a material advantage in strength.

Trench load tests at Iowa State College, under W. A. Schlick, were completed on 20-in. pit-cast pipe. Some of these tests were made with no internal pressure and others with a pressure as high as 1400 lb. per sq. in. Reports on these offer a method to determine the allowable supporting strength of pipe subjected to internal water pressure.

The work at Cornell University, under the direction of E. W. Schoder, involved tests on friction loss through various bends and pipe fittings. These tests will bring out the effects on friction loss of the radius of curvature of the inside corners of tees and crosses.

Tests on organic coatings and linings were made under the direction of S. R. Church.—*A. S. T. M. Bulletin.*

Plate Glass Windows

Plate glass insurance companies recommend from time to time that a radiator shield be interposed between the radiator and the plate glass window in order to protect it from breaking. This procedure is employed more in climates where there are quick changes in outside temperature. With a sudden drop in temperature and a quick response by the fireman, breakage of plate glass windows may occur.

The radiators are usually quite close to the winows when breakage occurs, usually a foot to eighteen inches. In most cases when the distance is greater than this there is usually no danger although it would seem advisable to take the precaution of using shields if there is any suspicion at all that the windows may be affected by the heat.

Conventions and Expositions

American Society of Mechanical Engineers: Meeting at Hartford, Conn., June 1-3. Secretary, Calvin Rice, 33 W. 39th St., New York City.

National District Heating Association: Annual Convention, June 2-5; Statler Hotel, Boston, Mass. Secretary, D. L. Gaskill, 603 Broadway, Greenville, O.

American Society of Heating and Ventilating Engineers: Semi-annual meeting, June 22-25; Swampscott, Mass. Secretary, A. V. Hutchinson, 51 Madison Ave., New York City.

American Society for Testing Materials: Annual meeting, June 22-26; The Hotel Stevens, Chicago. Secretary-Treasurer, C. L. Warwick, 1315 Spruce st., Philadelphia, Pa.

National Association of Power Engineers: Annual convention and exposition, September 7-11; Convention Hall, Kansas City, Mo. Secretary, F. W. Raven, 417 S. Dearborn st., Chicago.

Southern Power and Machinery Show: October 19-24; Textile Hall, Greenville, South Carolina.

Third International Conference on Bituminous Coal: November 16-21; Carnegie Institute of Technology, Pittsburgh, Pa. Chairman, Thomas S. Baker, President, Carnegie Institute of Technology.

Recent Trade Literature

Alloys: The American Brass Company, Waterbury, Conn.; four-page news bulletin containing numerous items showing the application of a corrosion resisting alloy in many industries, for ducts, fans, tanks, kettles, etc.

Alloys: The International Nickel Company, Inc., 67 Wall st., New York City; 12-page reprint of a technical paper on a corrosion and heat resistant nickel-copper-chromium cast iron. Mentioned among applications are pipe, valves, fittings, boiler specialties, impellers, propellers, oil burners, etc.

Alloy Steel Castings: The Duriron Company, Inc., Dayton, Ohio; eight-page catalog giving corrosion resistance data and physical properties of alloy steel castings for resistance to corrosion, heat and pressure. Included is information on machining, welding and heat treating.

Boiler Feeders: McDonnell & Miller, Wrigley Bldg., Chicago; four-page announcement regarding boiler feeders for all sizes of boilers, hand or gas fired, oil burner or stoker. An emergency switch to stop an oil burner or stoker if water level gets unduly low is featured.

Electric Immersion Heaters: H. O. Swoboda, Inc., 3400 Forbes st., Pittsburgh, Pa.; four-page leaflet discussing the heating of asphalts, tars, compounds, oils, varnishes, etc., with electric heaters.

Fans: Coppus Engineering Corporation, 344 Park ave., Worcester, Mass.; eight-page, well illustrated bulletin of fans of two types for man-cooling. Characteristics, performance and uses are considered.

Fans: B. F. Sturtevant Company, Hyde Park, Boston, Mass.; 12-page bulletin announcing propeller fans of pressed steel and including a discussion of the features, specifications, prices, dimensions and information on safety guards, pent houses, automatic shutters, and wall mounting panels.

Fans: Westinghouse Electric and Manufacturing Company, Mansfield, Ohio; 20-page dealer catalog devoted to ceiling, kitchen exhaust, marine, railway coach, desk, and bracket fans. Engineering information and prices included.

Floats: W. H. Nicholson & Company, 12 Oregon st., Wilkes-Barre, Pa.; four-page data sheet on welded steel floats for traps, tanks, water columns, level controls, etc.

Generator Units: Westinghouse Electric & Manufacturing Company, East Pittsburgh, Pa.; 12-page publication describing units where a small amount of energy is required but with no central station supply available. Belt-driven generators, turbine-generator units, portable gas engine driven units and electric service plants are included, with specifications and a section on applications.

Humidifiers: The Bahnson Company, Winston-Salem, N. C.; 16-page booklet describing automatically-controlled humidifiers for plants and buildings, showing a typical lay-out, and including general information on the importance of relative humidity.

Insulation: National Development Bureau, Department of the Interior, Ottawa, Canada; a 25-page mimeographed report on the building insulation industry in Canada, 1930, and the low temperature heat insulation industry in Canada, 1929.

Pressure Regulators: Spence Engineering Company, Inc., 110 E. 42nd st., New York City; loose-leaf data book containing a set of bulletins describing completely several types of pressure regulators, solenoid operated regulators, temperature regulators, automatic expansion valves, strainers, etc. Numerous illustrations, charts, etc., are included.

Unit Ventilators: B. F. Sturtevant Company, Hyde Park, Boston, Mass.; 48-page thoroughly illustrated catalog of non-recirculating, recirculating and partial recirculating unit ventilators (equipped with air filters) for use in schools, club houses, public buildings, hospitals, churches, offices, etc. Capacity tables, drawings of arrangements and specifications are included.